INTERNAL COMBUSTION ENGINES

INTERNAL COMBUSTION ENGINES

THEIR THEORY, CONSTRUCTION AND OPERATION

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PREFACE

THE intention of the authors in the preparation of this book has been to present in as simple terms as possible the fundamental and theoretical principles relating to the internal combustion engine, and to describe the various methods of applying these principles to practical construction. The book does not in any way treat of the proportioning and the strength of the various machine parts.

The general treatment of the subject is indicated by the various chapter headings. Thus the first five chapters relate to definitions and theoretical considerations, the subjects being as follows:

In the discussion on theoretical cycles in Chapter III, very little reference has been made to cycles not in actual use. The cycles are considered principally with reference to their practical application and any danger of confusing the mind of the student by a multiplicity of theoretical cycles of no practical value is avoided. The main idea of Chapter IV is to show how the lines of the real cycles differ from those of the theoretical cycles laid down in the previous chapter, and to discuss briefly the reasons for such difference.

The five chapters following, VI to X inclusive, take up the phenomena of combustion, the various gas-engine fuels, and the formation and properties of the fuel mixture. Thus, Chapter VI treats of combustion in general and discusses the most important properties of the gases usually found in gas-engine practice. Tables are given embodying the important constants for many of

these gases, and the chapter ends with some type computations on fuel mixture and exhaust gas constants which may serve as a guide for similar work by the student.

The question of gas-engine fuels is treated in the next three chapters by dividing all the fuels into three classes: the solid, the liquid and the gas fuels. Chapters VII and VIII take up the first two of these classes. Broadly speaking, neither class of fuel is directly available for gas-engine work, hence it was thought desirable to show in these chapters also the means by which these fuels are rendered available. Accordingly, Chapter VII discusses producer gas and describes the construction and operation of the most important types of producers, while Chapter VIII consists largely of a description of the various vaporizing devices used for crude oil, gasoline, kerosene, and alcohol. The latter fuel, recognizing its growing importance, has been treated in some detail. Chapter IX relates to industrial gases, pointing out briefly the method of their manufacture and giving the most important gas constants.

In Chapter X, after discussing the fuel mixture and its most important properties, an attempt has been made to collaborate the most important experiments on the variation of the specific heat of the fuel gases with a view to ascertaining the present state of our knowledge on this point.

Chapter XI gives a brief outline of the historical development of the gas engine, the various types being described only where they are of importance in connection with the development of modern forms. This is followed in Chapter XII by an extended description of the most important forms of internal combustion engines found in the market at the present time. The aim of this chapter has been not only to show how the various manufacturers have solved what is fundamentally the same problem, but also to familiarize the student, by means of a large number of illustrations, with the main constructive features of the gas engines at the disposal of the man requiring power.

Two of the important problems connected with the gas engine are the questions of ignition and of governing. The former is taken up in detail in Chapter XIII. This chapter, after mentioning briefly other methods of ignition, concerns itself mainly with ignition by electric spark, as being the most important method

used to-day. The chapter also takes up briefly two other gas-engine auxiliaries: mufflers and starting apparatus.

Chapter XIV treats the governing problem by discussing first the principles of the various systems of governing employed, and afterward shows the mechanical details of the governors used. It is beyond the scope of this book, however, to treat of the principles of governor design.

Chapter XIV discusses various methods in use for determining the necessary cylinder dimensions of a gas engine to develop a certain given power, or conversely, to determine the probable power for a given engine. The most reliable of these methods appears to be based upon the necessary charge volume for the given power. This method, the authors believe, was originally due to Güldner, and was adapted from that author's hand-book "Entwerfen und Berechnen der Verbrennungsmotoren."

For the determination of the power of automobile engines, two additional semi-empirical formulæ are given, and the results of the computations by the various methods are compared by means of a type example.

The remaining three chapters of the book treat of what might be called the economic side of the internal combustion engine.

Chapter XVI takes up the methods used in the testing of gas engines. The rules followed in this country will be of course the code laid down by the American Society of Mechanical Engineers. The Code of the German Society of Engineers, however, is appended, because it gives additional information upon some of the points involved and because it treats in greater detail of gas producers.

The results of tests on engines and producers are discussed in Chapter XVII. The various factors affecting economy are taken up in somewhat greater detail than has been done in any of the previous chapters. Tables are given showing the results of numerous tests and these should prove valuable in furnishing a guide as to what may be expected of other installations in the matter of fuel economy.

Finally, while Chapter XVII treats only of the question of fuel economy, Chapter XVIII takes up the entire financial problem relating to the gas engine. It shows in brief, as far as the information is available, the probable capital cost of the installation,

the cost of erection, the operating expenses, etc. It is shown most strongly that the question of fuel cost is not always the important item of the problem, a point which is often lost sight of in discussions regarding the comparative merits of various prime movers. It must be confessed that reliable information regarding some of the factors in this financial problem is still very scarce, owing, of course, directly to the comparative youth of the gas engine. It should also be pointed out that many so-called comparisons between various prime movers, as between steam and gas, are often based upon hypothetical assumptions that fit only the particular case under discussion, and any generalization of the results obtained often leads to serious misrepresentation.

The book is of course largely compiled from different sources and is in the main an outgrowth of a course of lectures on the internal combustion engine delivered to students of Sibley College for the past three years. Acknowledgments are due to numerous writers upon the subject for the facts and statements presented. The acknowledgments have generally been made in the body of the book, but the authors desire to extend herewith special acknowledgment to the following European authorities: Messrs. H. Güldner, Dugald Clerk, Bryan Donkin, and Aimé Witz, and to Professor C. E. Lucke of Columbia University and Mr. F. E. Junge of New York.

Thanks are also due to various manufacturers for kindly furnishing the text and illustrations of Chapter XII.

Ithaca, N. Y.,
April 23, 1908.

PREFACE TO THE SECOND EDITION

In presenting the second edition of this book the authors wish to express their obligation to all those who kindly aided in pointing out errors, and especially to Professor H. P. de Schweinitz of Lehigh University, for a thorough revision of the first two chapters.

February 1, 1909.

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CHAPTER I

INTRODUCTION, DEFINITIONS, CLASSIFICATION, AND FORM OF INTERNAL COMBUSTION ENGINES, INDICATED AND BRAKE HORSE-POWER

1. **Mechanical Work.** — Work is done when resistance is overcome; it is measured by the product of the resisting force and the distance through which that force is moved. If one pound is lifted one foot high in opposition to the force of gravity, a quantity of work, measured by the product of one pound by one foot, is performed, which quantity is known as a *foot-pound,* and is the unit of measurement for mechanical work in countries where the pound is a unit of weight and the foot a unit of distance. If 20 pounds are lifted 15 feet the work performed would be similarly 20 × 15 foot-pounds = 300 foot-pounds.

In countries where the metric system is used mechanical work is measured by the product of the resisting force in kilograms (2.2046 pounds) multiplied by the distance in meters (3.2808 feet); the product is expressed in *kilogrammeters* (7.233 foot-pounds).

The foot-pound or kilogrammeter is a gravity measure which depends on the intensity of the force of gravity at the place, and varies with that force. The variation is, however, so slight for different positions on the earth's surface that for all practical engineering work no sensible error is produced by considering it a constant quantity.

The unit of measurement usually employed by engineers for expressing *the rate of work,* or the quantity of work done in a given time as one second or one minute, is the *horse-power,* which has been arbitrarily defined as equivalent to 550 foot-pounds per second or 33,000 foot-pounds per minute. This quantity is considerably greater than the power a horse can exert, at least for any considerable length of time; it was first

used by James Watt in defining the power of the steam engine and has been established by long use as a definite measure of power. In France the term *Force de Cheval* is applied to a rate of work of 75 kilogrammeters per second ($542\frac{1}{2}$ ft. lbs.) or 4500 kilogrammeters per minute (32549 ft. lbs.).

In general, if W be the work performed against the pressure or resisting force p while moving through the space or volume v,

$$W = pv. \tag{1}$$

Work is done when force is applied so as to produce motion in the direction of action of the force, and also when force is employed in changing the velocity of a body already in motion. The latter condition is of considerable practical importance and can be considered as follows: Suppose a body whose mass is M be moving in a certain direction with a velocity u, and let a force exerting a momentum P be applied in the direction of motion, required to find the effect produced by this force acting through the small time t, during which the body moves through the distance v, and has at the end of the time the velocity u'.

The momentum produced by the force in one unit of time is P, and in t units of time it is Pt. Since this is equal to the increase of momentum produced, we have

$$Pt = M\,(u' - u).$$

As the distance is equal to the mean velocity multiplied by the time, we have

$$v = \tfrac{1}{2}\,(u' + u)\,t.$$

By multiplying the above equations,

$$Pvt = \tfrac{1}{2}\,(Mu'^2 - Mu^2)\,t;$$

dividing by t,

$$Pv = \tfrac{1}{2}\,(Mu'^2 - Mu^2) \tag{2}$$

Pv is the mechanical work done in overcoming a resistance; the expression $\frac{1}{2}\,Mu^2$ is the kinetic energy. From this it is seen that the mechanical work done is measured by the increase in the kinetic energy produced.

The mechanical work done by a fluid during a change of volume from v to v' is equal to the mean resistance overcome, or

pressure exerted, p, multiplied by the change of volume. That is, in general,

$$W = \int_v^{v'} pdv \tag{3}$$
$$W = p\,(v' - v)$$

In the operation of an engine, the working fluid expands and contracts as the piston moves forward and backward, and in one or more revolutions returns to its initial condition, so far as pressure, volume and temperature are concerned, and then passes through the same stages of expansion and contraction as before. The period through which these changes take place is termed a *cycle.*

The work performed in a cycle would be equal to the mean pressure, p, exerted, multiplied by the total volume, v', swept through; that is

$$W = pv'. \tag{4}$$

Mechanical work can be represented by a diagram in which the pressure exerted or resistance overcome, p, is represented by the ordinates, and the volume v by the abscissa. Such a diagram is called a *pressure-volume diagram;* its area is equal to $\int pdv$, and is proportional to the work performed.

Thus in Fig. 1–1, if the distances parallel to OY represent the pressure at any given point, and the distances parallel to OX the corresponding volume, then will the total work done in changing from the highest to the lowest pressure and from least to greatest volume be represented by the area of the figure $a\ b\ d\ e\ f$.

2. Heat. — Heat is a peculiar form of energy; it may be generated by the application of mechanical work, the amount so produced being exactly proportional to the mechanical energy which disappears. Conversely, mechanical work may be done by the action of heat, and for every foot-pound of work so done a definite amount of heat is put out of existence. Heat is also produced by a form of chemical action known as *combustion,* during which operation fuels are burned.

3. Temperature. — The *temperature* of a body is defined by Maxwell* as "its thermal state with reference to its power of communicating heat to other bodies."

* Theory of Heat.

A body transmitting heat to another is at a higher temperature and is said to be hotter; conversely, one receiving heat is at a lower temperature and is said to be colder.

Heat flows from a hotter to a colder body, but not conversely,

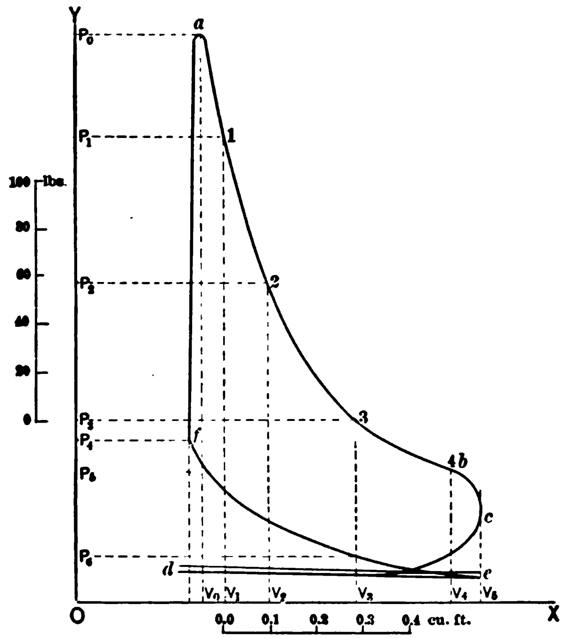


Fig. 1-1. — Pressure-volume or Work Diagram.

and the rate of flow increases with the difference of temperature, although probably not exactly in the same ratio. The difference of temperature thus causes a flow of heat in a manner somewhat smiliar to that caused by a difference of pressure in the case of water.

The terms *hotter* and *colder* are relative ones commonly applied to distinguish substances having relatively a higher or lower temperature. It should be noted that temperature is that property of heat which refers to its intensity or transmission power also, that heat energy may exist at different temperatures, and, furthermore, in one condition may be much colder than in another.

The following scales of temperatures are in common use in which the temperatures, of freezing and boiling water under a barometric pressure of 29.92 inches are taken as points of reference.

The Centigrade scale was introduced by Celsius, professor of astronomy in the University of Upsala about 1742; in it the freezing-point is marked 0 degrees and called zero, and the boiling-point is marked 100 degrees. The simplicity of dividing the distance between the points of reference into 100 parts and calling each of them a degree has caused it to be generally adopted along with the Metric System for scientific use, especially on the Continent of Europe. The other scales are called by the names of those who introduced them.

Fahrenheit of Dantzig, about 1714, introduced a thermometer scale in which the freezing-point was marked 32 degrees and the boiling-point 212 degrees, the space between the reference points being divided into 180 equal parts called degrees, and the graduation extended above and below the points of reference. A point 32 degrees below freezing was called zero. Despite the inconvenience of the scale of the Fahrenheit thermometer it is in general use by English-speaking people for commercial and business purposes, and for that reason will be used principally in this treatise.

Réaumur introduced a thermometer scale about 1730 in which the freezing-point is marked 0 degrees and the boiling-point 80 degrees, which is used to some extent on the Continent of Europe for medical purposes.

The following table gives the comparative value of the three thermometric scales:

THERMOMETRIC SCALES

	Fahrenheit	Centigrade	Réaumur
Degrees between freezing and boiling..	180	100	80
Assumed temperatures at freezing-point	32	0	0
Assumed temperatures at boiling-point	212	100	80
Comparative length of a degree.......	1	1.80	2.25
Comparative length of a degree.......	$\frac{5}{9}$	1	$\frac{5}{4}$
To transform into absolute temperature add	460°	273°	218°

4. **Absolute Temperature** is an expression for the value of temperature measured from an ideal point called the *absolute zero*, which is assumed to be the lowest possible point on any scale of temperature. The position of the absolute zero can be calculated by the law of expansion of a perfect gas, which is expressed by the simple equation

$$\frac{pv}{T} = R \tag{5}$$

in which p = the pressure, v = the volume of a given mass of gas, T = the absolute temperature, and R = a constant which varies only with the different kinds of gas. This equation can be considered as the characteristic equation of a permanent gas, from which T can be computed if p, v, and R are known, which is the case with most of the gases.

It is evident that the value of one degree of absolute temperature can be taken at pleasure as equal either to that on the Centigrade or Fahrenheit scale.

The exact location of the position of absolute zero is somewhat in doubt since it is determined by the relative expansion of air, nitrogen, or hydrogen under a constant pressure; these gases are not perfect gases and the expansion in volume per degree of increase in temperature may not be exactly the same as for a gas which could not be liquefied for any conditions of pressure or temperature. Preston in his work on the Theory of Heat states that the most trustworthy observations indicate that the absolute temperature of freezing water is 273.14 Centigrade, which would correspond to 491.65 Fahrenheit. It is sufficiently near for all practical purposes to consider the temperature of freezing water on the absolute scale as 273 degrees Centigrade or 492 degrees Fahrenheit, and these numbers will be used in this treatise in reducing to the absolute scale.

From this it is seen that to reduce to the absolute scale it is necessary to add to the temperature, if expressed in degrees, Fahrenheit 460, or if in degrees Centigrade 273.

5. **Thermometers.** — Instruments for measuring temperature are called *Thermometers*.

The expansion of a gaseous, liquid or solid body under con-

stant pressure is almost, if not exactly, proportional to the increase of temperature, estimated from absolute zero.

In the thermometer in common use the temperature is measured by the expansion of mercury, confined in a glass tube, from which the air has been exhausted. Such a thermometer is quite satisfactory within the range of temperature through which mercury will remain liquid. In the better grades of mercurial thermometers the graduations are cut with extreme care directly on the stem. The glass is carefully selected and is permitted to season or age until molecular changes have stopped before graduation. The general appearance of such thermometers is shown in Fig. 1-2. When thermometers are likely to be used in temperatures which would send the mercury column above the limits of the graduations, it is desirable to have an extra bulb, called a *safety-bulb,* at the top, to prevent breaking from overheating. Mercurial thermometers can be used from a temperature about 40 degrees below zero to 600 degrees above zero Fahrenheit. By filling the space above the mercury with some neutral gas as N or CO_2 under pressure the upper limit may be raised some hundred degrees; but as the melting-point of glass is low, the upper limit can scarcely ever exceed 800 degrees to 900 degrees Fahrenheit.

FIG. 1-2.—Mercurial Thermometer.

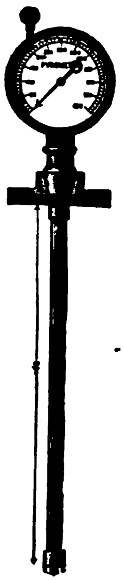


FIG. 1-3.—Metallic Pyrometer.

METALLIC THERMOMETERS in which the expansion of a metal, or the difference in expansion of metals of two different kinds, is multiplied by a system of levers so as to move a hand over a dial are frequently used for the measurement of temperature. Such thermometers are sometimes called *pyrometers.* An illustration of such a thermometer is shown in Fig. 1-3.

The metallic thermometer can be used for temperatures not exceeding 1200 degrees to 1500 degrees Fahrenheit, but it is sel-

dom an instrument of accuracy and is extremely liable to accident. The scale of these instruments should be frequently compared with the boiling-point, and adjusted if not found correct.

It has already been shown that air or permanent gases like nitrogen and hydrogen when under a constant pressure will expand in volume in proportion to the absolute temperature, or when confined so as to have a constant volume will increase in pressure in proportion to the absolute temperature.

It follows from this that if air be maintained at a constant volume and heated, its absolute pressure will increase with the absolute temperature, or *vice versa*, if it be maintained at constant pressure, its volume will vary with absolute temperature.

The air thermometer constructed in accordance with either principle is used as a standard way of measuring temperature, but because of the extreme difficulty of maintaining constant pressures or constant volumes it is an awkward instrument to use and is employed very little in the ordinary measurement of temperature. The Jolly form of constant volume air thermometer is shown in Fig. 1-4. The leg $C F$ has a flexible connection at the bottom and may be raised to maintain a constant volume of air from the bulb L to the line $A B$. The increase of pressure is measured by a scale attached to $C G$.

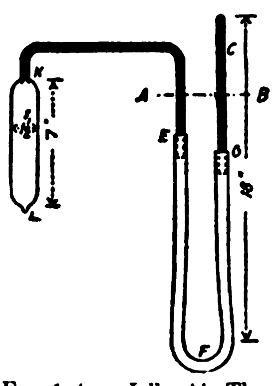


Fig. 1-4. — Jolly Air Thermometer.

Electrical Thermometers. — Temperature may be measured by electrical thermometers, of which there are two classes. In one class a conducting circuit is formed of two different metals, such a construction being frequently termed a *thermo-element*, a number of these connected together is known as a *thermopile*. In this construction an electro-motive force is produced which is proportional to the difference of temperature of the junctions and may be measured by a sensitive galvanometer. If one of the junctions be maintained at a constant or known temperature, the temperature of the other may be computed from the reading of the galvanometer. For the measurement of

high temperature, metals having a high melting-point, such as platinum and a platinum-iridium alloy, may be used for the elements.

The LeChatelier pyrometer is an instrument of this class;

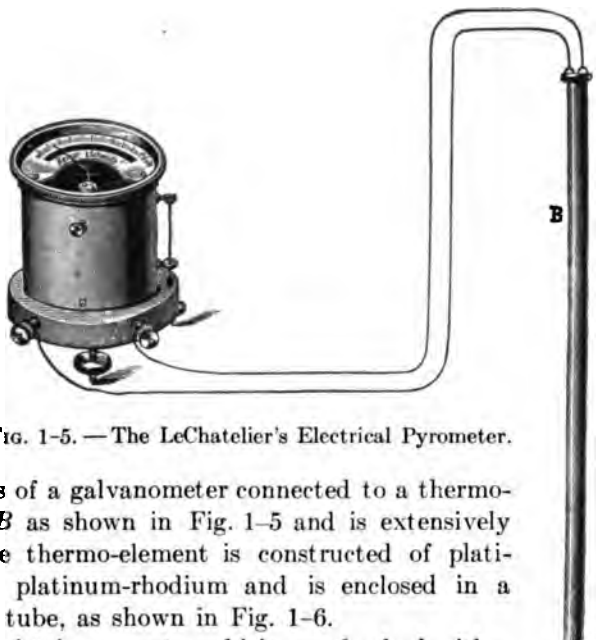


FIG. 1-5.—The LeChatelier's Electrical Pyrometer.

it consists of a galvanometer connected to a thermo-element *B* as shown in Fig. 1-5 and is extensively used; the thermo-element is constructed of platinum and platinum-rhodium and is enclosed in a porcelain tube, as shown in Fig. 1-6.

The Bristol pyrometer which may be had with a recording device belongs to the above class.

Another class of electrical thermometers is based on the law of increase of electric resistance of metals due to the rise of temperature. With this class of thermometers the difference in temperature can be determined from the measurement of the drop in potential for a known current passing through a coil. This method is employed in the platinum thermometers of Siemens, Calendar, and various others. The electrical thermometers are superior to all others for many uses.

OPTICAL PYROMETERS. — The approximate temperature of incandescent bodies may be determined by the color of the radiant rays. Pouillet, as the result of a large number of experiments, concluded that all incandescent bodies have a definite and fixed

color corresponding to each temperature, as shown in the following table:

Color	Temp. C.	Temp. F.
Faint red	525	927
Dark red	700	1292
Faint cherry	800	1482
Cherry	900	1652
Bright cherry	1000	1932
Dark orange	1160	2120
Bright orange	1200	2192
White heat	1300	2372
Bright white	1400	2552
Dazzling white	1500	2732

The fixed relation between color and temperature is due to the fact that the color of an incandescent body varies with the wave length which is a function of the temperature. A number of optical pyrometers have been devised which determine the temperature by the appearance of the heated body. The Mesuré and Noel pyrometer changes the wave lengths by the rotation of the plane of polarization of light passing through a quartz plate cut perpendicularly to its axis. In the use of the instrument the temperature is measured by noting the angle through which the analyzer is turned in order to produce a lemon yellow color.

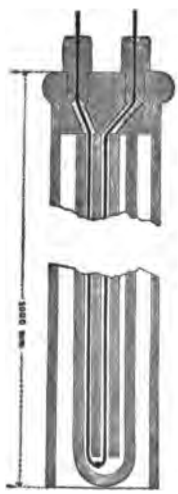


FIG. 1-6. — The Le Chatelier Thermo-Element.

The Morse thermo-gage, which is extensively used in the steel industry, consists of an incandescent lamp with a rheostat arranged so that the current flowing through it and its consequent brightness may be regulated. When the film of the incandescent lamp becomes the same color as that of the object the temperature is computed from the reading of a milli-voltmeter arranged to measure the current. The temperature corresponding to a given electrical reading is determined by calibration.

The optical pyrometers are convenient and of approximate

accuracy in determining high temperatures of incandescent bodies. They are of no value in determining temperatures of combustible objects.

VAPOR THERMOMETERS. — The pressure produced by a saturated vapor confined in a closed vessel increases with the temperature, in accordance with a known law or a law which may be determined. By providing a suitable pressure gage and attaching it to a closed vessel of the proper shape the temperature may be obtained from the pressure readings, the value of which are known, or from the dial readings of the pressure gage, which may be graduated by trial into degrees of temperature. An instrument of this type is made by Schaeffer and Budenberg and called by them a *Thalpotasimeter.*

CALOMETRIC THERMOMETER. — The temperature can also be measured by calometric methods, by heating a body of known weight and specific heat to the temperature which it is desired to measure, then transferring this body with as little cooling as possible to a vessel containing a known weight of water.

The equation for this operation will be expressed as follows:

$$W(t - t') = ws(t_x - t),$$

in which W = equivalent weight of water and its containing vessel.

w = weight of heated body.

s = specific heat of hot body.

t' = original temperature of water.

t = final temperature of water.

t_x = temperature of hot body.

A ball of platinum, copper, porcelain, or burned fire clay answers nicely for the body to be heated.

FUSION THERMOMETERS. — The temperature can be approximately determined from the known melting-points of metallic bodies, on the principle that the temperature will be higher than the melting-point of a body that melts, and lower than the melting temperature of one that does not melt. In place of metallic bodies a series of fusible clay cones called "Seger cones," whose melting-points are known, are often employed in the same manner.

6. Specific Heat. — Different materials of the same weights have different capacities for absorbing heat for a corresponding change of temperature; thus one pound of water will absorb about nine times as much heat as one pound of wrought iron for the same change of temperature. This peculiar heat capacity of bodies compared with water is termed *specific heat,* which is usually defined as follows:

The specific heat of a body is the ratio of the quantity of heat required to raise that body one degree in temperature, to the quantity required to raise an equal weight of water at standard temperature one degree.

The specific heat of water is not quite constant, being nearly three fourths of one per cent higher at the boiling and freezing temperatures than at fifteen degrees Centigrade; this is shown by the following table from the book of Steam Tables by Professor Peabody:

SPECIFIC HEAT OF WATER

Centigrade	Fahrenheit	Specific Heat
0°—5°	32°—41°	1.0072
5 —10	41 —50	1.0044
15 —20	59 —68	1.00
20 —25	68 —77	0.9984
25 —30	77 —86	0.9948
30 —35	86 —95	0.9954
40 —45	104 —113	1.00
45 —155	113 —311	1.008
155 —200	311 —392	1.046

7. Specific Heat of a Permanent Gas. — In general the conditions under which the change of temperature occurs should be distinctly specified, for the temperature of a body may be varied by the mechanical work done in the compression or expansion which occurs. The change of volume due to increase of temperature is so small for solids and liquids that the external work in change of volume may be neglected, but such is not the case for gases. For this reason the conditions under which the heating of a gas takes place must be stated when referring to its specific heat, and it has become customary to speak of two spe-

cific heats in connection with any gas, namely, the specific heat at *constant volume*, and the specific heat under *constant pressure.* The former is the quantity of heat required to raise the temperature of a unit mass of the gas one degree when its volume is kept constant, and the latter the quantity of heat required to raise the temperature of a unit mass one degree when the pressure is kept constant, compared with that of a unit mass of water. In the first case the pressure increases while the volume is kept constant and no external work is done; in the latter the volume increases under constant pressure, and an amount of external work is done which is measured by the product of the pressure by the change of volume.

The specific heat of a gas at constant volume bears a known relation to the specific heat at constant pressure, so that if one be determined experimentally the other may be computed. It will be shown later that the specific heat at constant pressure equals the specific heat at constant volume plus a constant which depends on the nature of the gas. That is,

$C_p = C_v + R$ when expressed in heat units, and
$K_p = K_v + JR$ when expressed in foot-pounds.

The following table gives the specific heats of constant pressure and volume of the principal gases:

TABLE OF SPECIFIC HEATS

Gas	C_p	C_v	γ	Gas	C_p	C_v	γ
H	3.4090	2.4177	1.41	CO	0.2479	0.1738	1.41
O	0.2175	0.1543	1.41	CO_2	0.2169	0.157	1.29
N	0.2438	0.1729	1.41	CH_4	0.5929	....	.....
Air	0.2375	0.1684	1.41	C_2H_4	0.4040	0.320	1.26
H_2O (vapor)	0.4805	0.3585	1.34				

In the above table C_p = Specific heat of constant pressure.
C_v = Specific heat of constant volume.
$\gamma = C_p \div C_v$.

The specific heat of a perfect gas is considered constant by most English writers who discuss the subject. Quite a number

of experiments have shown, however, that it varies with both the pressure and the temperature.

In the French work by A. Witz the following formula is given for the change of specific heat with pressure:

$C_p = a + b\,(p - 1)$, in which a and b are constants for each gas and p the pressure in kilos per square centimeter.

The specific heat of all vapors which are liquefied at moderate temperature undoubtedly increases with the increase of temperature. This subject is fully discussed in Chapter X of this work.

In this work the specific heat of gases will be considered constant unless otherwise mentioned. Any error caused by such consideration will not usually be serious, and by so doing the various formulas which express the heat capacities under actual working conditions are much simplified.

The specific heat of a mixture of various gases constituting a known weight or mass is equal to the mean specific heat of the mixture, and is found by multiplying the weight of each component part by its specific heat and dividing the sum of the products by the total weight.

8. The Heat Unit. — Heat is measured by its capacity to raise the temperature of a known weight of water. The unit of measurement is termed a *calorie* in the metric system, and a *British thermal unit* (B. T. U.) in the English system. A calorie is commonly defined as the heat required to raise one kilogram of water from the freezing-point to one degree Centigrade, and a British thermal unit (B. T. U) that required to raise one pound of water from 32 to 33 degrees Fahrenheit. Because of the variation in the specific heat of water near the freezing-point Professor Peabody in his Tables of Saturated Steam defines the thermal unit as that required to raise the temperature from 62 to 63 degrees Fahrenheit, or from about 15 to 16 degrees Centigrade.

The specific heat of water changes slightly for different temperatures as already noted, but it can be considered as constant without sensible error for all ordinary purposes in the measurement of heat when the temperature is maintained between the freezing and boiling points.

It is noted from the above statement that heat is measured not by the temperature alone but by its ability to heat a mass of water from one temperature to another; the total heat expressed

in thermal units being equal to the product of the weight of water by the change of temperature. Thus if 20 pounds of water be heated 25 degrees Fahrenheit, the heat required is the product of 20 times 25 = 500 B. T. U.

9. Mechanical Equivalent of Heat. — The mechanical equivalent of heat is the amount of work expressed in mechanical units which may be performed by the transformation of one heat unit into mechanical work. The value of the mechanical equivalent of heat was determined experimentally by Joule who found that 772 foot-pounds was equivalent to one B. T. U. or 425.6 kilogrammeters to one calorie. The determinations made later and with more accurate instruments by Rowland, reduced to the sea level and to 45 degrees of latitude, give the following values which are now generally adopted:

Expressed in calories, $J = 426.9$ kilogrammeters.
Expressed in B. T. U., $J = 778$ foot-pounds.

In this work J is used as the symbol for the mechanical equivalent of heat, and A as the reciprocal of J. That is, $J = 1/A$.

The experiment by means of which the equivalent value of the heat unit was determined in units of mechanical work serve to prove the general mechanical principle of the conservation of energy, which had been previously stated by Clausius as follows: "*In all cases where work is produced by heat, the quantity of heat consumed is proportional to the work done; and conversely, by the expenditure of the same amount of work the same quantity of heat may be produced.*" This principle is often called the first law of Thermodynamics.

10. Entropy. — One of the qualities or properties of heat which cannot be measured by any simple physical apparatus is termed "Entropy." This same quality was named by Rankine "the Thermodynamic Function"; its value is such that its change during a given time multiplied by the absolute temperature equals the total heat which may be transformed into mechanical work. From this definition it is noted that the product of absolute temperature by change of entropy is the measure of the capacity of heat for performing mechanical work.

As an illustration, if a gaseous body under pressure be allowed

to expand without receiving or giving off heat its entropy would remain constant and any mechanical work performed would be done at the expense of the heat existing in the body. A change which takes place without gain or loss of heat is termed *adiabatic*, which condition corresponds to that of constant entropy.

Expansion or compression of a body taking place without change of temperature is called *isothermal* expansion or compression, and lines drawn in a diagram indicating this property are termed "isothermal" lines.

Change of entropy with respect to heat is in many respects analogous to change of volume in respect to mechanical work; it has already been shown that the mechanical work is equal to the change in volume multiplied by the resistance or pressure overcome. If we denote the total heat capable of being transformed into mechanical work by Q, the change of entropy by ϕ and the absolute temperature by T, we shall have the following equations:

$$W = (v - v')\,p.$$
$$Q = (\phi - \phi')\,T.$$

From the first of the above equations it is noted that no mechanical work can be done without a change in volume, and further, that the amount of work done is measured by the change of volume multiplied by the *pressure*.

From the second it is seen that no change in the amount of heat which a body contains can take place without a change in entropy, for when $\phi = \phi'$, $Q = O$. The amount of heat transferred is measured by the change of entropy multiplied by the absolute temperature.

From the above

$$v - v' = \frac{W}{p}$$

From which is seen that the change in volume is equal to the mechanical work performed divided by the mean pressure.

$$\phi - \phi' = \frac{Q}{T}$$

From which it is seen that the change in entropy is equal to the total heat transformed into work divided by the absolute temperature.

11. **Classification of Engines** — The action of an engine is, in general, to produce motion against a resistance or to perform work. Engines are popularly classified in accordance with the nature of the working fluid, as hydraulic engines, steam engines, gas engines, oil engines, etc. They may be more scientifically classified in accordance with the nature of the working process as pressure engines and heat engines. In the *pressure engine* work is produced by change of pressure without change of temperature, as illustrated in the piston water engine. In the *heat engine* work is produced by transforming heat into mechanical work, which process is accompanied with change of temperature, and usually, also, a change of pressure.

In mechanical structure engines are of two classes, *reciprocating* and *rotary*. In the *reciprocating engine* a piston free to move in a cylinder is pushed backward and forward by alternate changes in pressure of the fluid against either face. The reciprocating motion of the piston is converted into a continuous rotary motion by a mechanism usually consisting of crank and fly-wheel which will be described later. In the *rotary engine* a rotary motion is directly produced by the force due to pressure, impulse, or reaction acting upon revolving blades or pistons arranged in a suitable casing. The term is often confined to a structure with a revolving piston in which motion is produced by a difference of pressure, whereas the term *turbine* is applied to the structure when rotation is produced by impulse or reaction of the jet. In a general way the turbine is a species of rotary engine. This treatise will be principally confined to internal combustion engines having a piston with reciprocating motion.

Engines are classified as *single acting* when the propelling force is applied to one side of the piston only, and as *double acting* when it is applied alternately to both sides.

Engines are classified as *simple, compound, triple, expansion,* etc., depending on the number of cylinders through which the working fluid passes in succession as it expands from highest to lowest pressure.

12. **Classification of Heat Engines.** — Heat available for use in a heat engine is usually produced by a species of chemical action termed *combustion.* Heat engines may be classified in accordance with the location of the place of combustion with

respect to the working cylinder as *external combustion engines* and *internal combustion engines.*

The external combustion engines include steam and other vapor engines, hot air engines, and some forms of gas or oil engines; the internal combustion engines include all the usual forms of gas and oil engines in which the fuel is consumed in the working cylinder. In this work the term *gas engine* will be frequently used as including all forms of internal combustion engines adapted to burn gas or vapor irrespective of the nature of the fuel.

Of these various types of engines the steam and the gas engines are the only ones of practical commercial importance at the present time. The hot air engine was built extensively about fifty years ago, and its theory was thoroughly investigated at that time. It failed as a commercial machine because of the high cost of repairs and operation; it is principally useful at the present time as illustrating the practical application of certain thermodynamical principles.

As the steam engine and hot air engine have a practical bearing on the internal combustion engine, a short description is inserted.

13. The Steam Engine. — The mechanism of the steam engine and its mode of operation should be familiar to all students of the internal combustion engine. The term steam engine is used here in its broad sense, including the boiler, the engine proper, and all the accessories necessary for its operation. In the mode of operation of the steam engine, steam is produced at any desired pressure by the combustion of fuel in a furnace beneath the boiler, which latter is a strong closed vessel containing a certain amount of water and into which water is introduced by a feed pump as desired. The steam engine proper is provided with a cylinder in which is fitted a piston which is propelled by the steam pressure acting on one or both sides. The admission and discharge of the steam are controlled by a valve or valves moved by the mechanism of the engine, the form of which varies greatly with different types. In the more common form, a slide valve is used which is propelled backward and forward by a valve rod moved either by an eccentric or by a short crank attached to the main shaft. The valve is operated so as to admit steam at nearly boiler pressure back of the piston for a portion of the stroke, and

then to cut off communication with the boiler, after which the piston is pushed forward by the expansive force of the steam. The valve is moved at the end of the stroke so as to open communication between the cylinder and the exhaust pipe, which in the case of a non-condensing engine discharges into the atmosphere and in case of a condensing engine discharges into a more or less perfect vacuum.

The motion of the piston is communicated, by means of a piston rod which slides through a stuffing-box at the end of the cylinder, to a block called a *cross-head* which moves in guides, from which motion is communicated by a connecting rod to the crank of the main shaft so as to produce rotary motion.

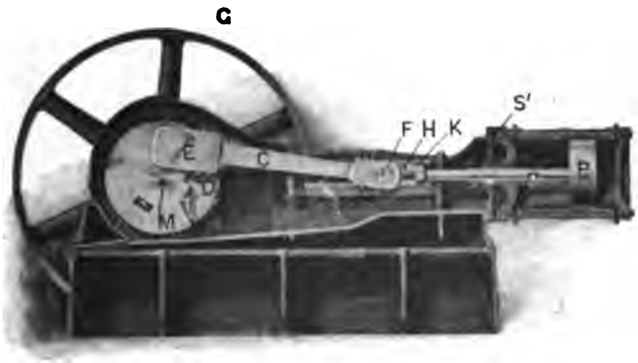


FIG. 1-7. — Double Acting Steam Engine.

The train of mechanism of the ordinary double-acting steam engine which is used to communicate motion from the piston to the fly-wheel is shown in Fig. 1-7, in which P represents the piston, p the piston rod, H the cross-head, C the connecting rod, M E the crank, E the crank pin, F the wrist pin, M the main shaft, G the fly-wheel, S the stuffing-box, which is used to prevent leakage of steam around the piston rod. The cross-head moves in guides K which direct its motion. One end of the connecting rod E has a circular motion, the other a rectilinear motion. In the view referred to the valves which admit steam alternately to the ends of the cylinder are not shown.

A view of a two-cylinder single-acting engine is shown in

Fig. 1–8 in section. In this engine the steam is admitted only at one end of the cylinder at the proper intervals of time by the sliding motion of the piston valve *V*, which is operated by the

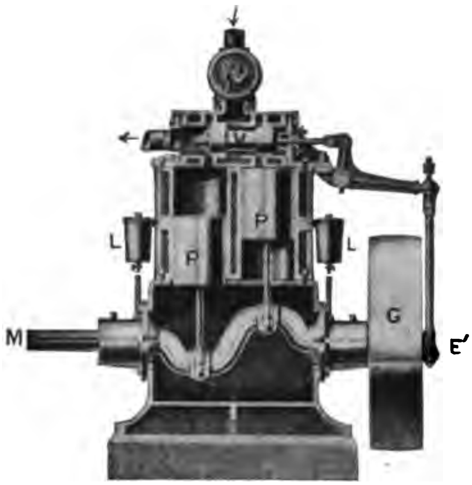


FIG. 1–8. — Two-cylinder Single-acting Steam Engine.

mechanism of the engine by means of a small crank attached to the main shaft and the various link connections shown.

Steam engines are frequently built compound, in which case the steam works in succession in a small and large cylinder, termed respectively the high-pressure and low-pressure cylinder. These cylinders may be arranged side by side as in the view of the single-acting engine, Fig. 1–8, or they may be arranged in tandem as shown in Fig. 1–9. In the tandem compound engine shown

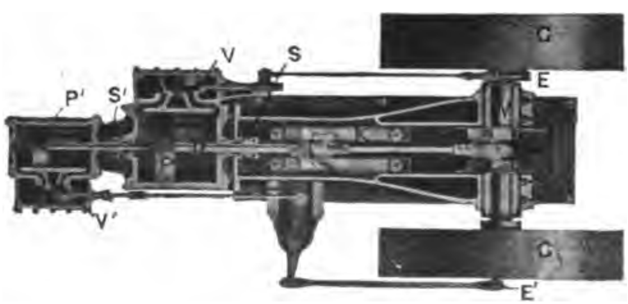


FIG. 1–9. — Tandem Compound Steam Engine.

the supply of steam is admitted, to the high-pressure cylinder by the slide valve V' operated from the small crank E', and to the low-pressure cylinder by the slide valve V operated from the eccentric E.

Since the application of pressure to the piston as described above is periodic and not constant, a fly-wheel consisting of a heavy mass of metal must be applied as shown, to produce uniform motion. To regulate the speed a governor is used, which is constructed so that the variation in centrifugal force either tends to cut off the supply of steam or to close the steam valves if the speed become higher than desired, or the reverse.

An indicator or pressure volume diagram of a steam engine is shown in Fig. 1–35. In this diagram vertical distances are proportional to pressures per square inch acting on the piston, and horizontal distances to the space passed through by the piston. The diagram shows the relation of pressure and volume at any point. The work done on the piston is proportional to the area of the diagram.

The steam engine, as will be shown later, does not realize in the work performed as great an efficiency on the basis of the heat values of the fuels employed as the gas engine; but it has the advantage of being adapted for the burning of solid fuels under its boiler without converting the same into gas, and this in a measure sometimes compensates for the greater heat losses. The steam pressure is exerted on the piston for a much larger portion of the stroke than an explosion in a gas engine piston, and as a consequence the inertia of moving parts is depended upon less for uniform speed than in the gas engine, as will be afterwards shown.

In the past the steam engine has had a great advantage over the gas engine, due to the fact that it has been more reliable in operation and could produce a more nearly uniform motion. The development of the gas engine has, however, removed in a large measure such defects, and at the present time there is no great difference in respect to reliability and regulation on the part of these two classes of engines. The use of a producer in which gas can readily be made from solid fuels also equalizes any advantages which the steam engine has had from that source.

14. Hot Air Engines. — The hot air engine is principally of importance to-day for its scientific value. Its actual commercial use is confined to pumping small quantities of water under favorable conditions. It is of scientific interest because it is the only heat engine yet produced which represents almost perfectly the standard ideal cycle of Carnot, with which the operation of nearly all heat engines is compared.

In the hot air engine a mass of air is successively heated and cooled; during the time that it is heated either its volume or pressure increases, and during the time it is cooled the reverse operation takes place. Mechanical work is performed by the change of volume or pressure which is utilized for moving a piston.

The principal varieties of air engines may be classified by the following distinctive features: 1. Change of temperature at constant pressure. 2. Change of temperature at constant volumes. 3. Heat received and rejected at a pair of constant pressures.

Ericsson's engine, best known as the caloric engine, may be taken as an example of the first class. In this engine, air is admitted from the atmosphere to the compressing pump at the lowest working temperature, and compressed, the temperature being maintained constant by the action of some refrigerating apparatus. The air when compressed enters a receiver. It is then admitted to the working cylinder, being heated on its passage to the higher temperature, so that its volume is increased and the pressure remains constant under the movement of the piston, then expands with its temperature maintained constant at the higher limit, and is finally expelled into the atmosphere, giving up its heat to the regenerator, to be used in heating the volume of air next introduced.

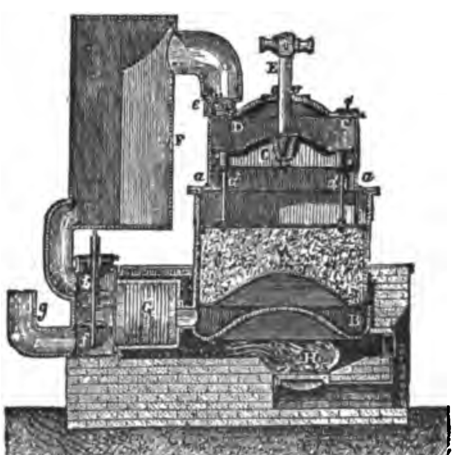


Fig. 1-10. — Ericsson Hot Air Engine.

This engine is represented in Fig. 1-10. *B* is a working cylinder, placed over the furnace *H*. This cylinder consists of two parts; the upper part in which the piston works is accurately turned, and the lower part in which the air receives heat from the furnace is less accurately made. *A* is the piston of the cylinder, consisting of an upper part which is accurately fitted and provided with metallic packings so as to work air-tight in the upper part of the cylinder. The lower part is somewhat smaller than the cylinder, is hollow, and filled with brick dust, fragments of fireclay, or some slow conductor of heat. The cover of the cylinder *B* has holes in it marked *a* to admit the external air to the space above the piston.

D is the compressing pump with piston, *C*, which is connected to the piston *A* by three or four piston rods, of which two are shown at *d d*. The space between the piston *C* and *A* is open to the external air. In operation the air is drawn into the compressing pump through the valve *c*, and is forced out after being compressed through the valve *e* into a receiver marked *F*. It is admitted at proper intervals of time by the valve *b* into the working cylinder *B*, being heated in its passage by hot plates in the vessel *G*, termed the regenerator.

It is further heated while in the working cylinder by the heat from the furnace *H*, the effect of which is to increase the temperature and volume underneath the piston. The increase of volume drives the piston to the end of its stroke. The exhaust valve *f* is opened by a mechanism connected to the engine and the working gases are forced outward through the regenerator, *G*, and discharge into the atmosphere through the pipe *g*. The regenerator is a vessel nearly filled with metallic plates which are heated by the escaping gases and give up heat to the engine gases, thus reducing in large measure the heat wastes from the engine.

In some forms of the Ericsson engine the air entering the compressor through the valve *c* is taken from the exhaust opening *g;* with this arrangement the same mass of air is repeatedly warmed and cooled, the changes of temperature taking place at constant pressures.

Ericsson's caloric engine was employed to drive a ship across the Alantic in 1853. The ship was 250 ft. long, had paddle

wheels 32 ft. in diameter. On its first trial trip the ship made twelve knots an hour with the wind, burning six tons of fuel per day. On the second trial the maximum speed was nine knots. After this unfavorable circumstances came to light and in 1855 the engine was taken out and a steam engine substituted.

Extended accounts of the Ericsson and other hot-air engines will be found in Knight's Mechanical Dictionary, 1873, in Appleton's Cyclopedia of Mechanics, 1878, in Bourne's work on the Steam, Hot Air, and Gas Engine, and in Rankine's Steam Engine.

Figure 1-11 represents the reciprocating parts of an air engine of the class in which temperature is changed at constant volume. Such an engine was designed and built by Dr. Robert Stirling about 1850, but improved by various other inventors, and is still built and sold for pumping small quantities of water as redesigned by Rider.

In the figure *D C A B* is the air receiver or heating and cooling vessel, which is provided with a furnace underneath, not shown in the diagram; *G* is the working cylinder with the working piston *H*. The receiver and cylinder communicate freely through the passage *F*, which is open at all times when the engine is working. Within the receiver is an inner receiver or lining of a similar figure, which has its bottom pierced with many small holes shown with dotted lines in the cut. The annular space between the receiver and its lining extending along the side of the receiver contains the regenerator, which consists of a series of oblong strips of metal with narrow passages between them. The inner surface of the cylindrical part of the lining from *A A* to *C C* is turned and the plunger *E* is fitted nicely in this portion. The upper portion of the receiver *D D* is supplied with a horizontal coil of fine copper tube, through which a current of cold water is forced or it is jacketed with cold water, and is termed the refrigerator. It is thus noted that the bottom of the receiver is kept hot while the top is kept cold. The plunger *E* is con-

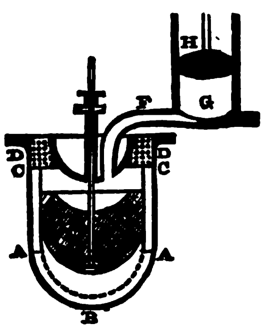


FIG. 1-11.—Stirling Hot Air Engine.

nected to the working mechanism of the engine so as to transfer a certain mass of air, which may be called the working air, from the hot to the cold end of the receiver, and in so doing making it pass up and down through the regenerator. The mechanism for moving the plunger *E* is so adjusted that the up-stroke of that plunger takes place when the piston, *H*, is at or near the beginning of its forward stroke, and the down stroke of the plunger when the piston *H* is at or near the beginning of the back stroke.

An air engine of class 3, which received and rejected heat at constant pressures, was designed by Jewell, but probably was never put into practical use.

Hot Air Engines Operated by Products of Combustion. — Another form of air engine, which Rankine in his Steam Engine terms a *furnace gas engine*, was first designed by Cayley of England and Barré of France, and was redesigned and improved and put on the market in this country about 1865 by Wilcox.

The engines of this class operate similar to a steam engine, pressure being produced by combustion in a closed furnace instead of in a steam boiler.

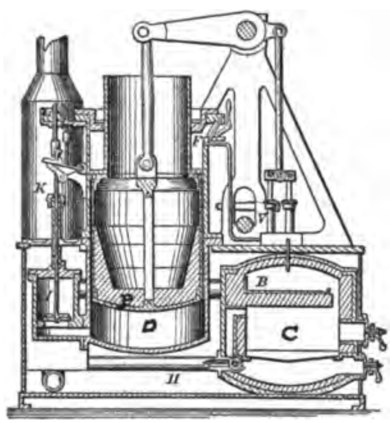


Fig. 1-12. — Wilcox Hot Air Engine.

In the operation of this engine (see Fig. 1-12) a pump draws air from the atmosphere through valve *F*, compresses it, and forces it into a strong air-tight furnace *C* through pipe *H*, where its oxygen combines with the fuel; then the hot gas produced by the *combustion* mixed with air is admitted into the working cylinder through pipe *B* and valve *I*, under pressure produced by the temperature of combustion, where it drives the piston *P* through part of its stroke at full pressure and through the remainder by expansion, until it falls to atmospheric pressure. It is discharged through an exhaust valve not shown and a pipe *X*. The entering air is heated by a regenerator which is warmed by the exhaust gases. In the

form shown (U. S. patent May 19, 1865) the furnace is fed with coal through a double valve, which is so constructed that it can be introduced without permitting the escape of more than a very small quantity of the compressed air. In the Wilcox engine, patented Sept. 19, 1865, the furnace was fed with petroleum oil under pressure.

Respecting Cayley's engine, Rankine states: "The cylinder, piston, and valves of this engine were found to be so rapidly destroyed by the intense heat and the dust from the fuel that no attempt was made to bring it into general practical use." Wilcox's engine never met with commercial success.

Most forms of the gas turbine belong to a class, in which pressure is produced by the heat of combustion of the gases before entering the turbine, either in a closed combustion chamber or in the inlet pipe. As the supply to a turbine is continuous no inlet or exhaust valves are required, and hence the trouble experienced in the early forms of this engine is greatly reduced.

15. **Structure and Mode of Operation of the Gas Engine.** — The mechanism of the ordinary gas engine, using the term in its general sense as covering all internal combustion engines, is similar in most respects to that of the steam engine, which has already been briefly described. It consists of a cylinder containing a piston which is moved by the pressure produced by the explosion of a charge consisting of a mixture of gas or vapor and air in the cylinder. The motion of the piston is communicated to a main shaft by a connecting rod similar to that used in the steam engine. The valve mechanism of the gas engine serves to admit and discharge the charge at the proper interval of time and is operated by the mechanism of the engine. In the early development of the gas engine a slide valve was used to a considerable extent, but at the present time the poppet valve is commonly used as it has been found to withstand high temperature better than the other form.

In order to start the gas engine some external force must be provided to introduce the combustible charge into the working cylinder and give it the initial compression. This may be done in the first instance by revolving the engine by extraneous power, which puts the piston in motion and serves to draw in the necessary gas and air by suction and to compress the same. After the

engine is in operation the inertia of the moving parts keeps it in motion and serves to draw in and compress the charge.

It is quite evident that the amount of work performed will be proportional to the mass or weight of the charge. For this reason it is desirable that the charge be under as much compression as practicable at the time of ignition, and all modern internal combustion engines provide means for compressing the charge previous to ignition either outside or inside of the working cylinder. In nearly every case, in the modern engine, the compression is completed in the working cylinder.

The principal events in the operation of an internal combustion engine are as follows:

1. Charging or suction, during which time the charge is drawn into the cylinder.

2. Compression, during which time the charge is compressed.

3. Ignition, explosion, and expansion, during which time heat is supplied which causes the combustion or explosion followed by the expansion due to increase of volume caused by motion of the piston. Ignition may take place under conditions of (1) *constant volume*, (2) *constant pressure*, or (3) *constant temperature*.

4. Exhaust, during which time the products of combustion leave the cylinder.

16. Classification of Internal Combustion Engines. — Gas engines are scientifically classified in accordance with the mode of applying heat during ignition as follows:

1. Engines receiving heat with charge at constant volume; these will be called in this work *Explosion Engines*.

2. Engines receiving heat with charge at constant pressure; these will be called in this work *Pressure Engines*.

3. Engines receiving heat with charge at constant temperature; these will be called in this work *Constant Temperature Engines*.

In the first of the above classes of engines the charge may be ignited with or without previous compression; consequently this class may be subdivided into *non-compression* and *compression engines*, the non-compression engine has entirely gone out of use because of its low efficiency and small capacity for a given size. Engines of any class may be either two or four stroke cycle engines as explained.

This classification is based on the characteristic equation expressing the relation between pressure, volume, and temperature of a given weight of a perfect gas, which, as will be shown later, is

$$\frac{pv}{T} = R$$

in which p = absolute pressure.
v = the volume.
T = absolute temperature.
R = a constant for any given gas.

It is evident that the heat may be received while any of the variables, pressure, volume, and temperature, remains constant, that is, at constant volume as in Class 1, constant pressure as in Class 2, or constant temperature as in Class 3.

Internal combustion engines are often unscientifically classified by the nature of the working fluid as gas engines, petrol engines, and oil engines. This classification gives no consideration to the fact that any of the above classes will operate with any of the fuels named. The term *gas engine* is frequently used in this work in its general sense, as applying to any form of internal combustion engine.

Engines Igniting at Constant Volume, or Explosion Engines. — In these engines, which are the ones commonly used, the various operations are performed in the order mentioned above in each working cycle of the engine. These engines may be divided into two classes accordingly as they perform these operations, in one end of the cylinder, (*a*) in four strokes or (*b*) in two strokes.

In this class of engines the combustion is practically instantaneous and of the nature of an explosion, taking place under normal conditions while the volume of gas remains constant, thus producing an extremely rapid rise of pressure.

(*a*) *Four-Stroke Cycle Engine.* — The internal combustion engine most commonly used ignites the charge while its volume remains stationary, and requires four strokes for one cycle of operation. For this reason it is known as the *four-stroke cycle* or *four-cycle engine.* This engine as ordinarily built is a single-acting engine with all the operations performed on one side of

the working piston. A diagram showing its general construction and mode of operation for each stroke is shown in Fig. 1–13. In the operation of this engine the charge is drawn in during the first out stroke of the piston, is compressed during the return stroke, is ignited with the piston stationary at the end of the stroke and with the volume of the charge constant. It expands during the next out stroke and is exhausted and expelled from the cylinder during the next in stroke.

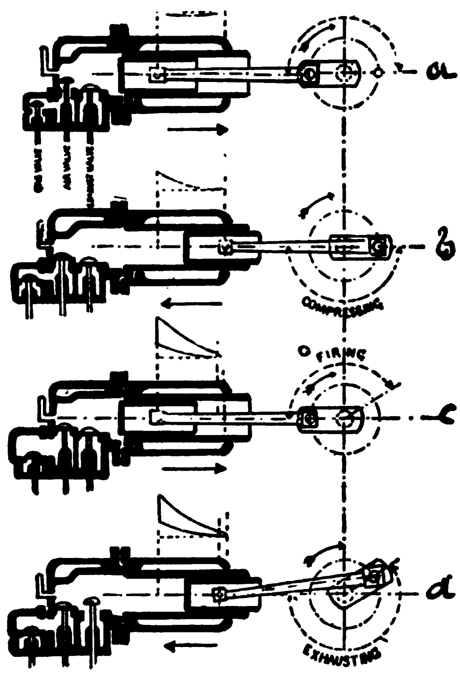


FIG. 1–13. — Diagram of Four-cycle Engine.

An engine of this kind was first described by Beau de Rochas in 1861, it was first built by Otto in 1876. The cycle on which it operates is for this reason often called the Beau de Rochas or Otto cycle.

(*b*) *Two-stroke Cycle Engine.* — An internal combustion engine, igniting as before, which is used for many purposes, is designed to perform the four operations above referred to in two strokes and is known as a *two-stroke cycle engine* or a *two-cycle engine.* A common form of such engine is shown in Fig. 1–14, in which form the engine is in part a double-acting engine and both sides of the piston constitute closed chambers. The crank case is made tight by the use of stuffing-boxes on the main shaft, and the suction operation is performed by the inward stroke of the piston which draws the charge into the crank case through the inlet *I*, where it is partially compressed by the out or return stroke of the piston and transferred through port *a*, which is uncovered at the proper time by the piston, to the ignition side of

the piston. At the same time the charge of previously burned gases is escaping through the port *E*. The compression is completed in the working cylinder *C*, after the piston has closed the transfer port *A*. Ignition occurs at the beginning of the out stroke and when the piston and volume in the working cylinder are stationary as in the preceding case. In the action of this engine suction takes place below the piston at the same time that compression takes place above, and compression takes place below the piston at the time of expansion above, as shown by the diagram of the crank circle in the figure. In certain large engines which are now made to operate on the two-stroke cycle system the suction and preliminary compression of the charge, which is performed in the engine just described by the working piston, is performed in a separate cylinder, the compression being completed in the working cylinder.

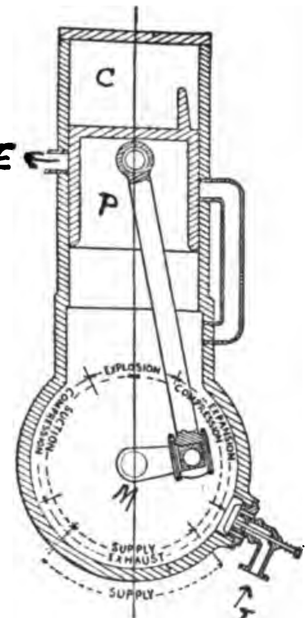


Fig. 1-14.—Two-cycle Engine.

The method of igniting the charge in common use will be described at length later in the book. In the class of engines referred to it consists of means for firing the charge instantaneously when under compression, and with its volume constant by an electric spark, a hot tube, or an open flame.

Engines Igniting the Charge at Constant Pressure.—The only engine of this class of practical importance is the Brayton, although under many conditions the Diesel engine ignites under constant pressure. The Brayton engine was at one time used extensively in America but is not now manufactured. In this engine the combustible and air for supporting combustion were supplied to the working cylinder under pressure which remained constant until the inlet valve closed; the combustion taking place during the admission of the air and com-

bustible. The work is performed by pressure acting during increase of volume in much the same manner as in the steam engine. In the Brayton engine, one form of which is shown in Fig. 1–15, the compression is performed in a compressor distinct from the power end of the working cylinder, and the heat is supplied at constant pressure. In the figure *B* is the working piston arranged to move in the cylinder *A*. The lower part of the cylinder *A* is the working cylinder, the upper part the air compressor which is arranged to deliver air into the reservoir *C*.

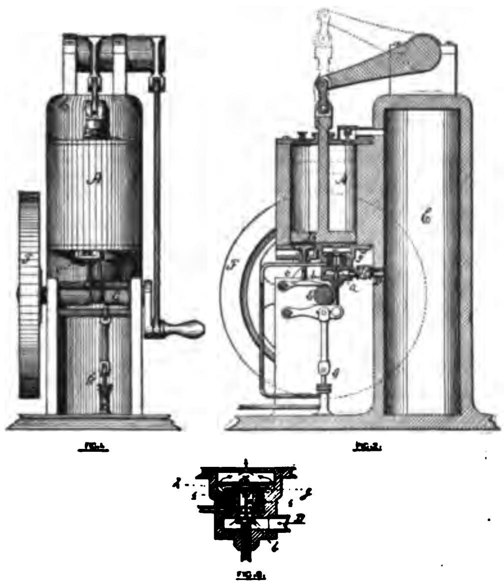


Fig. 1–15. — The Brayton Engine.

Oil is injected by the pump *G* into a vaporizing device (shown on a larger scale in 3 of Fig. 1-15) when it comes in contact with compressed air from the reservoir *C*. The inlet valve *b* and exhaust valve *c* are operated by suitable cams on the cam shaft *E*.

Engines Igniting the Charge at Constant Temperature. — Another class of engine, patented by Diesel, is arranged to supply the fuel during a portion of the working stroke at such a rate as to maintain the temperature constant, the working cylinder having previously been filled with air during

a suction stroke and compressed during a return stroke. In the Diesel engine the compression is sufficient to raise the air to a temperature high enough to ignite the fuel as it enters the working cylinder.

The Brayton engine as above described is a two-stroke cycle engine and the Diesel a four-stroke cycle engine, but both engines could be constructed to operate with either cycle.

17. **The Engine Indicator.** — This is an instrument designed to draw a diagram with ordinates proportional to the pressure which acts inside the cylinder at each point during the working and return stroke of the piston. It is briefly described here in order to give the student an idea of the method of obtaining the pressure volume diagrams which are frequently referred to in the work.

It consists essentially of (1) a part carrying a sheet of paper which is moved by proper mechanism in corresponding directions and proportional to the piston of the engine, and of (2) a part which carries a pencil which is moved a distance proportional to the pressure per square inch acting upon the piston.

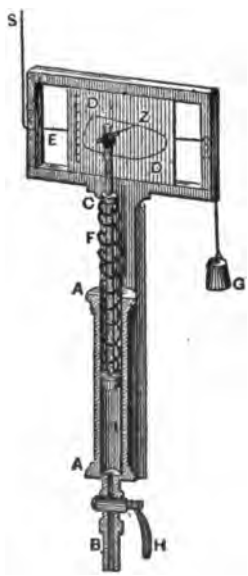


Fig. 1-16. — The Watt Engine Indicator.

The engine indicator was first designed by James Watt, substantially as shown in Fig. 1-16. This indicator was constructed with (1) a flat plate, *D B*, on which paper could be mounted which was moved in proportion to the motion of the engine piston, and (2) a cylinder *A A* which could be put in communication with the working cylinder of the engine by a three-way cock *H* and pipe *B*. In the cylinder *A A* was a piston whose motion was resisted by a spiral spring arranged to carry on its piston rod a pencil so constructed as to draw a diagram on the moving plate with ordinates proportional to the pressure.

The cock *H* can be turned so as to put the cylinder *A A* in communication with the air for the purpose of drawing a line showing the atmospheric pressure, which is called

the "atmospheric line." This simple form of indicator, although containing the essential elements of the modern indicator, was crude in construction and gave results which were far from accurate.

The modern indicator is an instrument of precision, and differs principally from that designed by Watt by the substitution (1) of an oscillating drum, called the paper drum, for the flat reciprocating plate, for carrying the paper, (2) of movable indicator springs in place of a fixed one, making it possible to regulate the length of the ordinates, and (3) a multiplying pencil motion in place of the direct one, whereby the motion of the pencil on the indicator drum is made greater than that of the indicator piston.

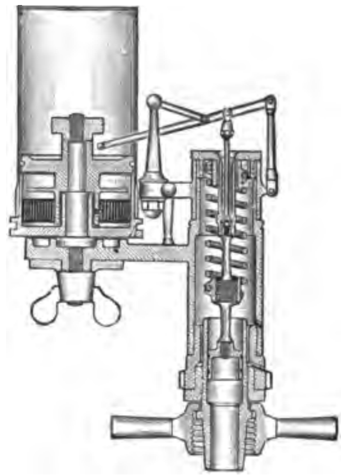


FIG. 1-17. — The Thompson Indicator.

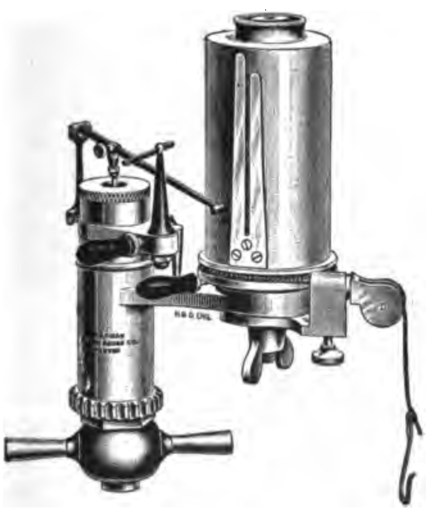


FIG. 1-18. — The Thompson Indicator.

A sectional and perspective view of the Thompson indicator, made by the American Steam Gauge & Valve Company, is

shown in Figs. 1–17 and 1–18. A perspective view of the Crosby indicator, which differs from the Thompson principally in the construction of the indicator spring and pencil motion, is shown in Fig. 1–19. For gas engine work the indicator spring is liable to be injured by heat. To lessen these difficulties most of the makers supply indicators with external springs, as shown in the attached view of the Tabor indicator, Fig. 1–20.

Fig. 1-19. — The Crosby Indicator.

The authors have found from an extensive experience in indicating gas engines that the indicator spring when arranged as in Fig. 1–17 can be kept from injury by surrounding the working cylinder with a water jacket or cup filled with water. The indicator cylinder is connected to the working cylinder by a pipe containing an indicator cock, which is arranged as in the Watt indicator to connect either with the air or the engine. The indicator drum is usually connected to some form of *reducing motion* by the indicator cord, which will move the surface of the drum proportional to the motion of the piston and not to exceed two or three inches, regardless of the stroke of the piston. Several forms of reducing motions with schemes for connecting will be shown in the chapter on the testing of gas engines, but will not be further referred to here.

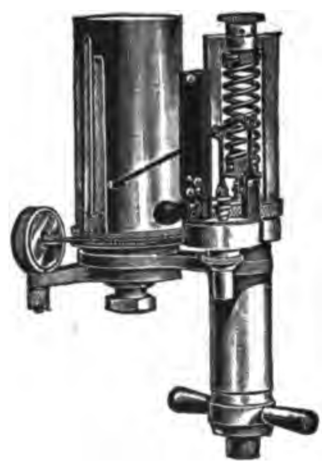


Fig. 1-20. — The Tabor Indicator with External Spring.

The class of engine indicator as described above is not adapted to take diagrams at extremely high speeds because of the inertia of the moving parts and because of the error due to stretching

of cords or flexible parts which connect the paper drum to the moving parts of the engine. For high speeds the optical indicator is preferable.

The optical indicator has been designed so that the only moving part is a small mirror which is arranged so as to project a ray of light on a ground glass screen or on a photographic plate.

Fig. 1-21.—Perspective View of Manograph.

The mirror is moved in one direction an amount proportional to the pressure acting on the piston of the engine, and in a direction at right angles an amount in proportion to the motion of the piston, so that the joint movement is proportional to the pressure volume diagram, the area of which represents the mechanical

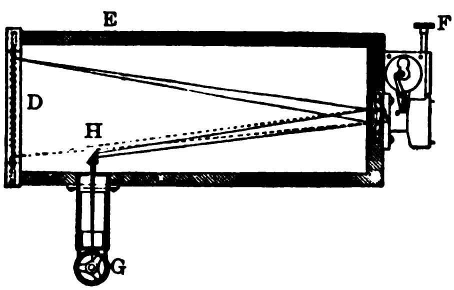


Fig. 1-22.—Horizontal Section of Manograph.

work performed by the working fluid in the engine cylinder. Such indicators are affected only to a slight degree by the rotative speed of the engine.

There are two instruments of this class on the market, one the *Manograph*, manufactured by J. Charpentier, Rue de Lambre, 20, Paris, the other the *Optische Indicator*, constructed by the

Elsaessische Electricitaets-Werke, Strassburg. A perspective view of the *Manograph* mounted on a tripod is shown in Fig. 1-21, a vertical section of same in Fig. 1-22; a detailed view of the mirror and engine connections are shown in Figs. 1-23 and 1-24.

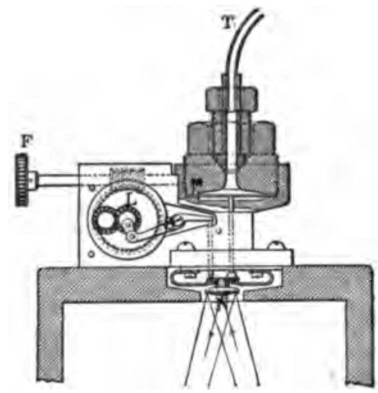


FIG. 1-23. — **Section of Manograph Mirror.**

The general construction is that of a photographic camera connected to the engine by tube T, and by flexible shaft R con-

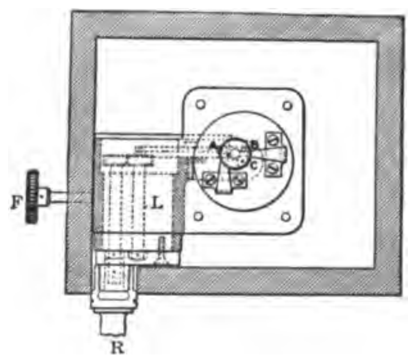


FIG. 1-24. — **Manograph Engine Connection.**

nected to the main shaft of the engine. An acetylene or electric lamp is located at G and its ray of light is projected through a perforated diaphragm to a prism, H, and thence to mirror N, at the back of the camera. The mirror N, supported on springs,

Fig. 1-23, is connected by a pin with the diaphragm M, which is in communication with the engine cylinder, so as to be tilted in one direction an amount proportional to the change of pressure in the engine cylinder. The mirror N is also tilted in a direction at right angles to the first motion by means of a crank with lever connection which is rotated from the shaft in proportion to the motion of the engine. The motion of the small crank can be set

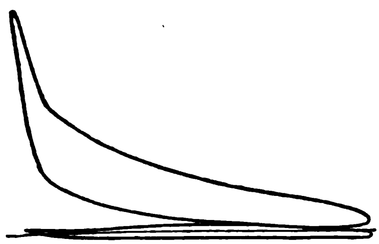


FIG. 1-25 — Diagram with Manograph.

in phase with that of the engine crank by the thumb screw F so that it will give a motion directly proportional to the piston of the engine. To make the errors as small as possible the angular motion of the mirror is made very small. The pressure scale of the manograph should be determined by carefully comparing the photographic diagram with a known pressure. The instrument

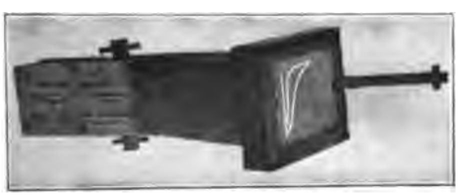


FIG. 1-26. — Optical Indicator.

is arranged to give a diagram which would have exactly correct proportions when the connecting rod has a length 4.5 times that of the crank, which is nearly the average proportions in actual practice and would make the resulting error small for other conditions.

Figure 1-25 represents a diagram taken with the Manograph from an engine making 1500 turns per minute and giving a maximum pressure corresponding to 158 pounds per square inch.

The Optische Indicator differs from the Manograph principally in details of construction. Its general appearance is shown in Fig. 1–26. A diagram taken from a motor operated with gasoline making 1000 turns per minute is shown in Fig. 1–27, the pressure scale of which is about 120 pounds per square inch.

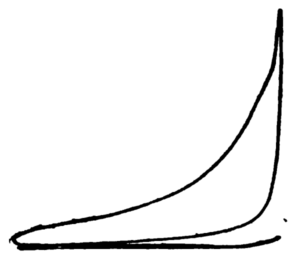


FIG. 1–27. — Diagram with Optical Indicator.

18. Indicated and Brake or Delivered Horse-Power. — The indicated horse-power which is generally denoted by the symbol I. H. P. is proportional to the area of the diagram obtained by use of the engine indicator, since this diagram has ordinates which are proportional to the pressures acting upon the engine piston at each point during the working and return strokes, and abscissa proportional to the corresponding space moved through by the engine piston.

The indicated horse-power (I. H. P.) is computed by use of the formula

$$\text{I. H. P.} = \frac{plan}{33000}$$

in which p = the mean effective pressure (m. e. p.) for the cycle of operation, acting on each square inch of the piston.

l = length of stroke in feet.

a = net area of piston in square inches.

n = number of cycles per minute.

33,000 = number of foot-pounds per minute in one horse-power.

The mean effective pressure (m. e. p.) is the mean ordinate

for all the strokes constituting the cycle multiplied by the proper pressure scale; it is best obtained by finding the net area by use of a *planimeter* (an instrument which will be described later), which is to be divided by the length of the diagram and multiplied by the scale of the indicator spring. The other quantities in the formulæ depend upon the dimensions and speed of the engine.

The *brake* or *dynametric horse-power,* for which the symbol in this work will be D. H. P., is that delivered from the main shaft of the engine and is consequently less than the indicated horse-power by an amount equal to the engine friction and internal losses.

The brake horse-power is usually measured by use of a special form of absorption dynamometer known as the Prony brake. Various forms of this brake have been employed, of which

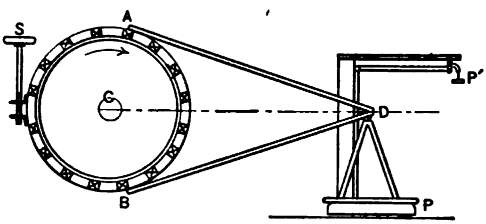


Fig. 1–28. — A Prony Brake.

two only are considered in this place. One form is shown in the diagram Fig. 1–28, which consists of a series of blocks connected by a leather strap or strip of iron and arranged so as to rub on the surface of a wheel attached to the main shaft. The brake is provided with two arms, the free end of which rests on a pair of scales. The amount of friction may be varied by use of a hand wheel or similar device as shown at *S*. The horizontal distance from the center of the wheel to the end of the arms is known as the arm of the brake and is denoted in the formulæ which follow by *a*. In the use of the brake the load is applied by turning the screw and is measured by the reading on the weighing scale. In a brake with the arm on one side only, as shown in the figure, the amount required to balance the overhanging brake arm must be deducted from the reading of the scales to give the net load.

The horse-power is calculated from the formula

$$\text{D. H. P.} = \frac{2\,\pi\,a\,n\,W}{33000}$$

in which n = number of revolutions per minute.
a = the brake arm in feet.
W = the net load on scales corrected for unbalanced effect of brake.

Another form of brake is shown in diagram, Fig. 1–29, which is convenient for testing small engines. It consists of a rope or strap which makes one or more turns around the wheel, the tension or pull on both ends of which must be known or measured. In the form shown a weight of known amount, w, is applied at one end, and a spring balance is employed to measure the resistance at the other end.

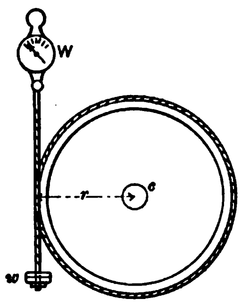


FIG. 1–29. — The Rope Brake.

The formula for this brake is as follows:

$$\text{D. H. P.} = \frac{2\,\pi\,r\,n\,(W-w)}{33000}$$

in which r = radius of the wheel in feet to center of strap.
W = the principal scale reading.
w = the lesser scale reading or weight carried.

In a modification of this brake the principal tension is received by a framework resting on a pair of scales, and the smaller resistance is absorbed by an upward pull on the platform of the same scales. With this arrangement the scale reading gives directly the difference of weights $W-w$.

In the use of the Prony brake heat is generated equivalent to the mechanical work absorbed. If the load is heavy it will be necessary to circulate water or some other heat-removing fluid inside of the rim of the revolving wheel or in some equivalent place.

19. **Forms of Indicator Diagrams.** — A few of the typical forms of indicator diagrams are considered in this place for the purpose of making the student familiar with the subject. They will be discussed at length later in the work.

Figure 1-30 is a hypothetical diagram of a four-cycle explosion engine with the events which take place on the various strokes marked. Thus the suction stroke is represented by *d e*, the compression by *e f*, the explosion by *f a*, the expansion by *a b*, the exhaust by *b c d*. The atmospheric line not clearly shown in the diagram would occupy a position intermediate between *c d*

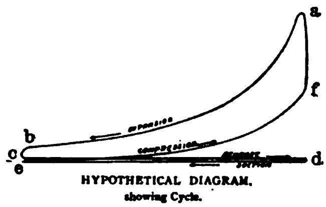


Fig. 1-30. — Four-cycle Engine.

and *e d*. The lines *c d* and *e d* on the indicator usually coincide with the atmospheric line for the reason that the spring used is too stiff to show such small variations in pressure as exist between the atmospheric line and the exhaust and suction lines.

Figure 1-31 shows diagrams of a two-cycle explosion engine in which the upper diagram, *a b c f*, is taken in the working cylinder, and the lower diagram, *e′ f′*, in the compressor. In this diagram the net work is the difference between that shown on the first diagram and that on the second.

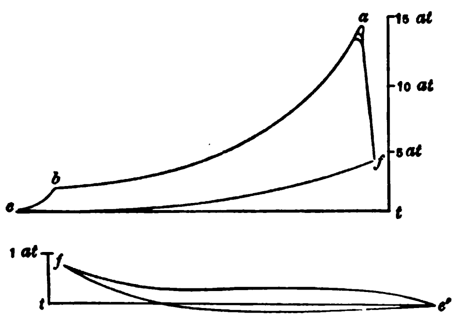


Fig. 1-31. — Two-cycle Engine Diagram.

Figures 1-32 and 1-33 are diagrams from a Brayton or constant pressure engine, Fig. 1-32 being taken from the working cylinder and Fig. 1-33 from the compression cylinder. It will be noted

from Fig. 1–32 that the pressure remains constant from *a* to *a′*, at which time communication is cut off from the compressor, after which the fluid expands from *a′* to *c* in much the same

FIG. 1–32. — Diagram from Brayton Working Cylinder.

manner as in the steam engine. The net work is proportional to the difference of the areas of the two diagrams.

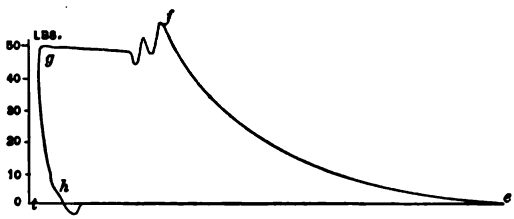


FIG. 1–33. — Diagram from Brayton Compressor.

Figure 1–34 is a diagram from a Diesel engine in which the temperature is supposed to be constant during that portion of the stroke represented by *a b*, during which time fuel is being supplied to the working cylinder.

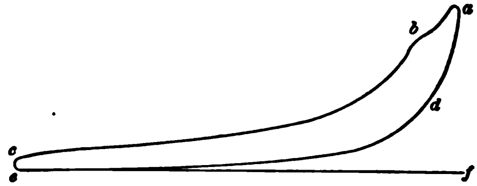


FIG. 1–34. — Diesel Engine Diagram.

Figure 1–35 is a diagram from a steam engine, the expansion line of which as referred to an hyperbola, *c e f g h*, which is asymptotic to the line of no pressure, *C D*, and of no volume, *C B*. Points in the hyperbola are obtained if the initial point, *c*, is

known, by drawing a vertical from c; then from C draw diagonals crossing $c\,b$ and $A\,B$. The intersection of a horizontal line from the intersection of the diagonal and $c\,b$, with a vertical line from

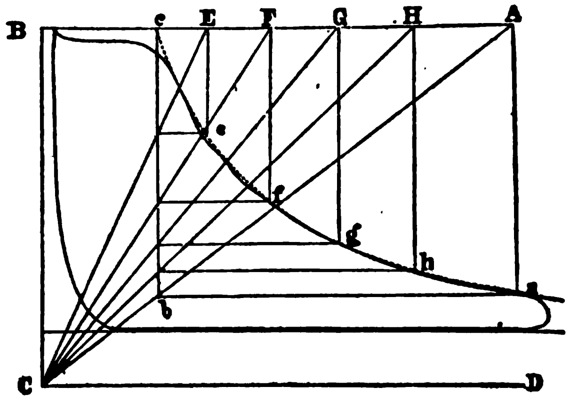


FIG. 1-35. — Method of Drawing Hyperbola.

the intersection of the same diagonal and the line $A\,B$, give points in the hyperbola.

Another method of drawing an hyperbola is shown in Fig. 1-36,

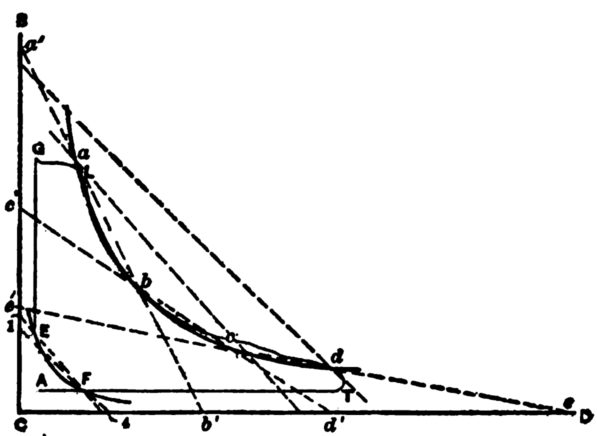


FIG. 1-36. — Method of Drawing an Hyperbola.

which represents an indicator diagram referred to lines of no volume and no pressure. This method is founded on the principle that the intercepts made by a straight line intersecting an

hyperbola and its asymptotes are equal. Beginning at any point as a, draw the straight line, $a' b'$, and lay off from the line $C D$, $b' b$ equal to $a' a$, then will b be a point in the hyperbola. Draw a similar line through b as $c' d'$ and find another point as c. Repeat the method until all the points required for drawing the curve are found.

The hyperbola is a useful line of reference in connection with indicator diagrams. As will be shown later, it represents the condition of isothermal expansion in the gas engine.

CHAPTER II

THERMODYNAMICS OF THE GAS ENGINE

1. **Notation.** — In order to comprehend the limitations of the actual engine it is necessary to understand how the working fluid behaves when subject to definite changes in an engine unaffected by friction or mechanical limitations, and we will for that reason give attention to the theoretical considerations relatting to the "internal combustion engine" which forms part of the science of Thermodynamics.

In the consideration of the theoretical action of a perfect engine the following symbols will be used:

A = the reciprocal of the mechanical equivalent of heat.
a = the absolute temperature of the freezing-point = 273 C. or 492 F.
C_p = the specific heat of constant pressure in heat units.
C_v = the specific heat of constant volume in heat units.
J = mechanical equivalent of heat = 778 in foot-pounds = 426 K. G. M.
K_p = specific heat of constant pressure in mechanical units.
K_v = the mechanical heat at constant volume in mechanical units.
p = absolute pressure for condition denoted by postscript.
p_0 = absolute pressure at freezing-point.
Q = the total heat of a given mass.
R = a constant for a given gas.
T = absolute temperature.
t = temperature Fahrenheit or Centigrade as marked.
v = volume of a given mass of gas, its condition being denoted by postscipt.
v_0 = the volume of a given mass of gas at freezing-point.
W = the mechanical work performed in mechanical units.
W' = the mechanical work in heat units.
α = the coefficient of expansion of a perfect gas = the reciprocal of a.
γ = specific heat of constant pressure divided by the specific heat of constant volume.

2. **Characteristics of Perfect Gases.** — In an internal combustion engine the work is produced by the change of volume and pressure of a gaseous mixture composed of atmospheric air

and the various products of combustion. This gas mixture, within the working limits of temperature, is obedient to the laws of the perfect gases, and for that reason in any theory of the internal combustion engine we are principally concerned with such laws and with the changes of volume and pressure in a perfect gas, in relation to its change of temperature.

By combining Boyle's law with Gay-Lussac's law it is learned that the product of pressure and volume of a given mass of a perfect gas varies directly as its absolute temperature, that is:

$$pv = p_0 v_0 \ (1 + \alpha t) \tag{1}$$

in which p equals the absolute pressure and v the volume of a given mass of gas at a temperature, t, above the freezing-point, p_o and v_o represent the pressure and volume of the same mass at the freezing-point, and α represents the coefficient of expansion of the gas per degree of absolute temperature. As has already been shown α will equal when expressed in the Centigrade system the reciprocal of 273 and when expressed in the Fahrenheit system the reciprocal of 492. If we denote the number of degrees between the freezing-point and absolute zero by a, then from the preceding explanation a will equal the reciprocal of α. If we denote the absolute temperature by T we shall have

$$T = a + t. \tag{2}$$

substituting $\frac{1}{a}$ for α in equation (1) we have

$$pv = \frac{p_0 v_0}{a} (a + t) = \frac{p_0 v_0}{a} T \tag{3}$$

in the above equation $\frac{p_0 v_0}{a}$ is a constant for each gas.

$$\text{Let } R = \frac{p_0 v_0}{a} \tag{4}$$

then we have

$$pv = RT. \tag{5}$$

The above equation may be considered the characteristic equation of a perfect gas, since it shows the relations between the pressure, volume and absolute temperature.

R is a constant which depends on the nature of the gas and

can be computed if the specific pressure, p_0, and specific volume, v_0 at standard pressures and temperatures are known.

It follows from the above that, $p_1v_1 = RT_1$, from which by comparing with (5)

$$pv : p_1v_1 :: T_1 : T \tag{5a}$$

The specific pressure p_0 is the weight of the atmosphere at the freezing-point under normal conditions; it is equivalent to that of a column of mercury 760 mm. high (29.921 inches). This reduced to pressure per unit of area is

$$p_0 = 10333 \text{ kilograms per square meter;}$$

or in English units,

2116.3

p_0 = ~~2146.32~~ pounds per square foot.
= 14.696 pounds per square inch.
= 29.921 inches of mercury.

The specific volume is determined from the density of the gas. The following table gives the specific volume in Metric and English measures for some of the more common gases:

VALUE OF v_0 AT LAT. 45°

	Cubic meters per Kilogram	Cubic feet per Pound
Air	0.7735327	12.3909
Nitrogen (N)	0.7963291	12.7561
Oxygen (O)	0.6996231	11.2070
Hydrogen (H)	1.116705	178.881
Carbonic Acid (CO_2)	0.5058741	8.10324

By substituting the values of p_0v_0, and a in the equation $R = \frac{p_0v_0}{a}$, the value of R can be found.

Thus for air, in French units

$$R = 10333 \times 0.77353 \div 273 = 29.20\ ;$$

in English units

$$R = 2116.3 \times 12.391 \div 492 = 53.22.$$

The following table gives values of R for a few gases:

	Values of R	
	English	Metric
Hydrogen (H)	770.3	422.68
Oxygen (O)	48.74	26.475
Carbon-dioxide (CO_2)	35.41	19.43
Air	53.22	29.20

The following table showing the specific heat of the ordinary gases is inserted here for convenience.

Table of Specific Heats

Name of Gas	Symbol	Specific Heat		$\frac{C_p}{C_v}$
		Constant Pressure C_p	Constant Volume C_v	γ
Air		0.2375	0.1684	1.406
Oxygen	O	0.2175	0.1552	1.403
Nitrogen	N	0.2438	0.1727	1.416
Hydrogen	H	3.4090	2.4110	1.414
Nitric oxide	NO	0.2317	0.1652	1.402
Carbonic oxide	CO	0.2450	0.1736	1.413
Carbon dioxide	CO_2	0.2000	0.1550	1.261
Steam	H_2O (saturated)	0.4805	0.3700	1.298
Methane	CH_4	0.5930	0.4680	
Acetylene	C_2H_2	0.3460	0.2700	
Disulphide carbon	CS_2	0.1569	0.1310	1.198
Olefiant gas	C_2H_4	0.4040	0.3330	1.125
Ammonia	NH_3	0.5084	0.3910	1.300
Alcohol	CH_2O_6	0.4534	0.4100	1.150

As before noted, the specific heat of gases increases with the temperature and possibly also with the pressure, which law will be referred to in discussing the application to special cases. The expanding products of combustion are composed of N, CO_2 H_2O, O, and possibly a trace of NO. For approximate computations the value of C_p/C_v for the burned gases may be taken at 1.37.

3. **General Relations of Heat Transmission to Changes of Volume and Pressure.** — When a quantity of heat dQ is supplied to a mass of gas it produces a complex result; the heat

warms the gas and raises its temperature, at the same time performing internal work by overcoming the molecular forces; it then develops external work, which is made apparent by the expansion of the gas against an external resistance. Denoting by dU the quantity of heat employed in warming the gas and in molecular work we will have

$$dQ = dU + Apdv,$$

in which pdv is the external work expressed in mechanical units and $Apdv$ its equivalent in heat units.

In the operation of a gas engine the mass of gas constituting the working substance expands during each cycle from one volume to another, passing through a series of successive changes of volume and pressure, and finally returns to its initial state. It is evident that when the mass returns to its initial state that the quantity represented by $\int dU$ is equal to zero. For this condition we shall have

$$dQ = Apdv.$$

Calling U the internal heat of the gas, which has already been shown to be a function of the volume pressure and temperature, we have

$$U = f\ (v.\ p.\ t.).$$

Since t can be determined from the values of p and v as indicated in equation 5, we can consider that in practice U is a function only of v and p and may write

$$U = f\ (v.\ p.).$$

The increase of internal heat for a variation of volume dv and of pressure $d\,p$ may be expressed as a total differential of the function, f, and we have

$$dU = \frac{\delta U}{\delta v}\,dv + \frac{\delta U}{\delta p}\,dp \tag{6}$$

By substituting the above value in the expression $d\,Q$ the thermal state of the gas may be represented as follows:

$$dQ = \frac{\delta U}{\delta v}\,dv + \frac{\delta U}{\delta p}\,dp + Apdv$$

or

$$dQ = \frac{\delta U}{\delta p}\,dp + \left(\frac{\delta U}{\delta v} + Ap\right)dv \tag{7}$$

It may be noted that the above equation cannot be integrated in its present form unless Q can be expressed as a function of the initial and final volumes and pressures of the gas. This demonstrates that the expenditure of heat required to make a gas pass from one state to another cannot be deduced from a knowledge of the extreme states, if one does not know the order and relation of the intermediate states.

The heat transferred at constant volume is equal to the specific heat C_v multiplied by the change of temperature. That is, $Q = C_v (T_1 - T)$, from which $dQ = C_v dt$. For a similar reason that transferred at constant pressure

$$Q = C_p (T_1 - T)$$
$$dQ = C_p dt.$$

For the condition of *transfer of heat at constant volume, dv* of equation (7) will equal 0, and equation (7) will become

$$dQ = \frac{\delta U}{\delta p} dp$$

Since the heat interchange for this case takes place at constant volume, it has been shown that

$$dQ = C_v dt.$$

By placing these two values of dQ equal we have

$$C_v \frac{dt}{dp} = \frac{\delta U}{\delta p} \tag{8}$$

from which the value of $\frac{\delta U}{\delta p}$ for heat interchanges at constant volume can be found.

For the *transfer of heat at constant pressure* the temperature changes take place without change of pressure, in which case dp of equation (7) = 0, and we have by substitution

$$dQ = \left(\frac{\delta U}{\delta v} + Ap\right) dv$$

Since the heat interchange for this case takes place at constant pressure, $dQ = C_p dt$, and by substitution

$$C_p \frac{dt}{dv} = \frac{\delta U}{\delta v} + Ap \tag{9}$$

from which the value of $\frac{\delta U}{\delta v}$ for heat transformation at constant pressure can be obtained. Substituting the values of eq. (8) and (9) in eq. (7) we have

$$dQ = C_v \frac{\delta t}{\delta p} dp + C_p \frac{\delta t}{\delta v} dv \tag{10}$$

which gives the value of the heat interchange for successive change of pressure and volume. This can be reduced as follows:

From eq. (5), $pv = RT$, hence $vdp = Rdt$ when v is constant, and $pdv = Rdt$ when p is constant. From this it follows that

$$\frac{\delta t}{\delta p} = \frac{v}{R}, \text{ and } \frac{\delta t}{\delta v} = \frac{p}{R}$$

Substituting the above values in eq. (10) we have

$$dQ = \frac{1}{R}(C_v v dp + C_p p dv) \tag{11}$$

Now, from eq. (5), we obtain by complete differentiation

$$pdv + vdp = Rdt, \text{ from which } vdp = Rdt - pdv$$

Substituting this in eq. (11) we may write

$$dQ = \frac{1}{R}\Big[C_v(Rdt - pdv) + C_p pdv\Big]$$

$$= C_v dt - C_v \frac{p}{R} dv + C_p \frac{p}{R} dv$$

But from eq. (5) $$\frac{p}{R} = \frac{T}{v}$$

hence finally $$dQ = C_v dt + (C_p - C_v)\frac{T}{v} dv \tag{11a}$$

4. Transformation to Different States. — The modes of transformation from one state to another depend upon the relation of p and v at different points, T being always determined, by the relation of p to v, from eq. (5).

For simplicity of treatment and for producing a standard for comparison it is assumed that the changes in the relations of volume, pressure, and temperature take place with one of the variables constant in the general equation (5a), p. 47.

Thus, if the volume remains constant during the change, we have $v = v_1$, and

$$\frac{p}{p_1} = \frac{T}{T_1} \tag{12}$$

which is the equation for constant-volume conditions. If the pressure remain constant, $p = p_1$ and

$$\frac{v}{v_1} = \frac{T}{T_1} \tag{13}$$

which is the equation for constant-pressure conditions. If the temperature remain constant, which latter is termed an *isothermal* condition, $T = T_1$.

$$\frac{p}{p_1} = \frac{v_1}{v} \tag{14}$$

which is the *equation of an isothermal line* for a perfect gas in a pressure-volume diagram.

Another standard of comparison is the transformation in pressure, volume or temperature which takes place without gain or loss of heat. This latter condition is called *adiabatic* and corresponds to that of constant *entropy*. It represents the conditions of the equation (11) when $dQ = 0$, in which case

$$C_v vdp + C_p pdv = 0$$

$$vdp + \frac{C_p}{C_v} pdv = 0$$

Substitute γ for $\frac{C_p}{C_v}$, then

$$vdp + \gamma pdv = 0.$$

which integrated between the limits p_1v_1 and pv gives

$$\log_e \frac{p}{p_1} = \gamma \log_e \left(\frac{v_1}{v}\right)$$

from which $\quad p_1v_1^\gamma = pv^\gamma = \text{constant} \qquad (15)$

which is the *equation of an adiabatic line* for a perfect gas in a pressure-volume diagram.

The equation for adiabatic transformation in terms of v and T can be obtained by substituting for p and p_1 the values as given in (5*a*) and reducing, which will give

$$Tv^{\gamma-1} = T_1v_1^{\gamma-1} \tag{16}$$

In a similar manner the adiabatic equation in terms of p and T can be obtained, which is as follows:

$$Tp^{\frac{1-\gamma}{\gamma}} = T_1p_1^{\frac{1-\gamma}{\gamma}} \tag{17}$$

5. Work Performed in Isothermal Expansion. — The work, W, performed when the gas expands isothermally from an initial volume, v to a volume v_1 can be calculated as follows:

The general formula for mechanical work is

$$W = \int p dv$$

but as $pv = p_1v_1$ for isothermal expansion

$$p = \frac{p_1v_1}{v}$$

$$W = p_1v_1 \int_v^{v_1} \frac{dv}{v} = p_1v_1 \log_e \frac{v_1}{v} \tag{18}$$

Since $pv = p_1v_1 = RT$ this may be written

$$W = RT \log_e \frac{v_1}{v} = pv \log_e \frac{v_1}{v}$$

The heat applied during isothermal expansion can be obtained by making $dt = 0$ and T a constant in (11a) and integrating. We will have

$$Q = (C_p - C_v)T \int_v^{v_1} \frac{dv}{v} = (C_p - C_v)T \log_e \frac{v_1}{v} = ART \log_e \frac{v_1}{v} \tag{19}$$

This value being the same as that of the external work indicates that *the heat applied during isothermal expansion is equivalent to the external work performed.*

It will be noted from (18) that an infinite increase in isothermal expansion will lead to an infinite amount of work. Thus in the equation

$$W = pv \log_e \frac{v_1}{v}$$

if v_1 be made equal to infinity the value of W also becomes infinite.

6. Work Performed in Adiabatic Expansion. — The work performed when the gas expands adiabatically from an initial volume, v_1 to a volume, v_2, can be found by substituting the value of $p = \frac{p_1v_1^\gamma}{v^\gamma}$ from formula (15) as follows:

$$W = \int_v^{v_1} p dv = p_1v_1^\gamma \int_v^{v_1} \frac{dv}{v^\gamma} = -\frac{p_1v_1^\gamma}{\gamma - 1}\left(\frac{1}{v_1^{\gamma-1}} - \frac{1}{v^{\gamma-1}}\right)$$

Therefore $$W = \frac{p_1 v_1}{\gamma - 1} \left\{ 1 - \left(\frac{v_1}{v}\right)^{\gamma - 1} \right\} \tag{20}$$

For infinite adiabatic expansion the work, W, does not become infinite as in isothermal expansion, since for this case $\frac{v_1}{v} = 0$, and

$$W = \frac{p_1 v_1}{\gamma - 1}$$

7. Relations of Heat to Entropy. — The heat transformations can be expressed as a function of the absolute temperature and entropy, which expression possesses some advantages for tracing heat interchanges over the pressure-volume equations.

For this case the variables in the equation become temperature, T, and entropy, ϕ, instead of v and p as in the preceding cases; in the diagram representing such conditions, horizontal lines would represent equal temperatures or isothermal conditions, while vertical lines would represent equal entropy or adiabatic conditions. The ordinates in such a diagram would then represent temperature, T, and the abscissa entropy, ϕ.

Since the heat interchange $d\,Q$ is equal to the product of the absolute temperature into the corresponding change of entropy, we have

$$dQ = Td\phi$$

$$d\phi = \frac{dQ}{T}$$

$$\phi - \phi_1 = \int \frac{dQ}{T}$$

For gases, if heat is supplied at constant pressure $dQ = C_p dt$

$$\therefore\ \phi - \phi_1 = \int \frac{C_p dt}{T} = C_p \log_e \frac{T}{T_1} \tag{21}$$

8. Carnot or Reversible Engine, Second Law of Thermodynamics. — A reversible engine is one that may be run in one direction so as to transform heat into work, or in the opposite direction so as to transform work into heat.

No actual heat engine is built in this manner, since such an hypothesis requires that all the gases exhausted shall pass through all the states in a reversed direction during compression and return to the initial state, which, because of the chemical

changes during combustion, is impossible in the internal combustion engine. The internal combustion engine can be considered as approximating the theoretical reversible engine, which thus becomes useful as a standard of comparison.

For the cycle of a reversible engine

$$\int \frac{dQ}{T} = 0$$

This is the highest attainable result with any heat engine, since it indicates that the heat transferred into work from motion in one direction would be returned to its source by an equal amount of work applied to drive the engine in an opposite direction. The above statement is *Carnot's principle*, which is often called the *Second Law of Thermodynamics*. It follows from this:*

(1) All reversible engines working between the same source of heat and refrigerator have equal efficiencies.

(2) The efficiency of a reversible engine is independent of the working substance.

(3) A self-acting machine cannot transfer heat from one body to another at a higher temperature.

It further follows from this that for any irreversible engine cycle the work for a given expenditure of heat is less than for a reversible engine; that is,

$$\int \frac{dQ}{T} = N$$

in which N represents the mechanical results of the work performed.

9. **Graphical Relations.** — The relations of the heat interchanges to the transformations of pressure, volume, and temperature will be more clearly understood by reference to a diagram. The pressure-volume diagram, which has for its ordinates lines corresponding to pressure and for its abscissæ distance corresponding to volumes, shows the conditions of uniform pressure by a horizontal line and of constant volume by a vertical line. On this diagram an isothermal line is represented by equation (14), $pv = p_1v_1$ which is the equation of an equilateral hyperbola of which the axes are the lines of zero volume and the line of zero pressure. Two methods of drawing the hyperbola have been

*Peabody's Thermodynamics, page 30

given in Art. 19, Chapter I. In the case of a steam engine it will be remembered that an isothermal condition is represented by an equal pressure line.

The *equation of an adiabatic line* on a pressure-volume diagram, as given in (15) is $\log \frac{p}{p_1} = \gamma \log \frac{v_1}{v}$. The values of the coordinates for drawing this curve can be found by assuming values of $\frac{v_1}{v}$ and finding the corresponding values, by use of a table of Naperian logarithms, of $\frac{p}{p_1}$. A table giving the values of γ for different gases has been given. It is usually assumed as 1.37 for gas engines and is subject to some correction for changes due to rise of temperature. The general relations of isothermal and adiabatic lines to pressure, volume and temperature is shown on the diagram Fig. 2-1.

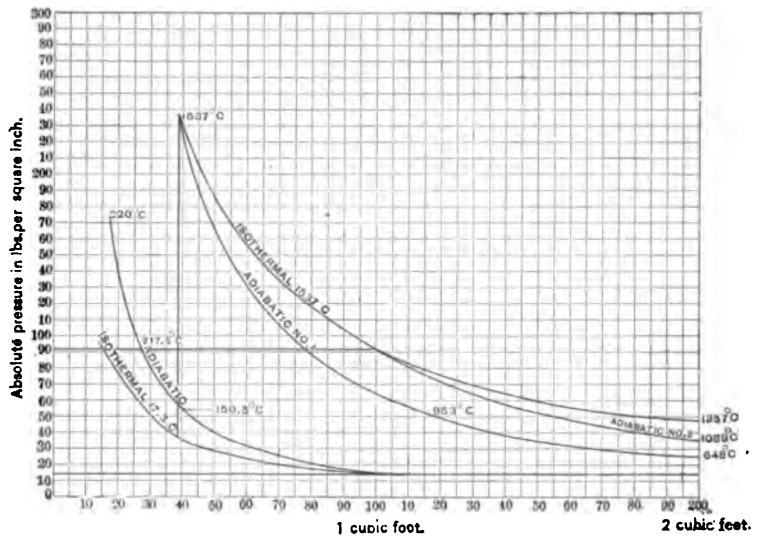


FIG. 2-1.—Relations of Isothermal and Adiabatic Curves.

Figure 2-1 shows isothermal and adiabatic expansion and compression lines drawn from the same points. This diagram shows that for a given number of expansions the adiabatic line falls

below the isothermal line, and also that the area between the adiabatic line and the base line is less than that between the isothermal and the base line. As this area represents the external work performed, it indicates that for a given number of expansions the work is greater in isothermal than in adiabatic expansion, which also follows from the demonstrations which have been given.

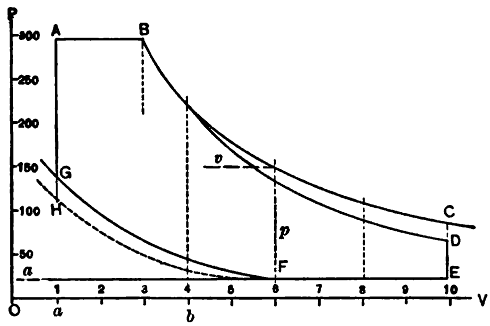


Fig. 2-2.— Isothermal and Adiabatic Changes Compared.

The various fundamental changes which may take place in a perfect heat engine are represented by the diagram, Fig. 2-2, in which *O V* is the base line from which pressures are measured and *O P* the zero volume line from which volumes are measured. In the diagram *G A* is a vertical line and represents the condition of receiving heat at constant volume; *A B*, a horizontal line, represents the condition of receiving heat at constant pressure; *B C*, a plain hyperbola, represents isothermal expansion, and *B D*, a logarithmic curve, represents adiabatic expansion; *D E*, a vertical line, represents the discharge of heat at constant volume; *E F*, a horizontal line, represents the discharge of heat at constant pressure; *F G*, a logarithmic curve, represents adiabatic compression, and *F H*, a plain hyperbola, represents isothermal compression. The area of the diagram, *A B D E F G*, represents the external work done with adiabatic expansion and compression, and *A B C E F H A* represents the work done with isothermal expansion and compression.

During the period of receiving heat at constant volume, represented by *AG*, the absolute temperature may be computed

at A, from (12), provided it is known at G, since it is proportional to the pressures at those points.

During the period of receiving heat at constant pressure, represented by $A\,B$, the absolute temperature increases in proportion to the volume, as shown in equation (13), and if known at one point may be computed at any other. As the volume is proportional to the distance from the line $O\,P$, the temperature on the line $A\,B$ will be proportional to that distance.

If the expansion is isothermal the temperature would remain constant from B to C. If the expansion is adiabatic, as from B to D, the temperature could be calculated from either equation (16) or (17) for various points of the curve.

The actual pressure-volume diagram as taken with the indicator shows lines which only approximate those for constant pressure, constant temperature, or constant volume as shown in Fig. 2–2. The corners of actual diagrams are likely to be rounded to a considerable extent and many of the transformations indicated may not appear.

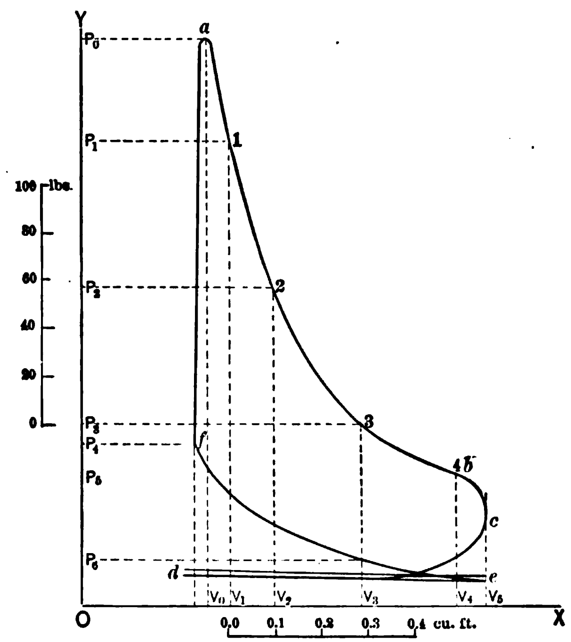


FIG. 2–3. — Four-cycle Engine Diagram.

Figure 2–3 represents a diagram of a 4-cycle engine in which the ordinates have been enlarged relatively with respect to the abscissæ and on which there have been drawn a line of no volume, often called the clearance line, and a line of zero pressure. The scale of ordinates is attached to the diagram. If we assume that the line *a f* is a vertical line and that the temperature at the lower end of that line is 567 degrees absolute, then we find by computation, as explained above, that for the point *a* it is 1995 degrees absolute. As absolute temperature *F* is 460 degrees higher than that shown on a thermometer, the temperature *F* at these two points would correspond to 107 degrees and 1535 degrees.

The heat transformations may also be represented by the entropy temperature diagram, in which case the ordinates become temperature and entropy instead of pressure and volume.

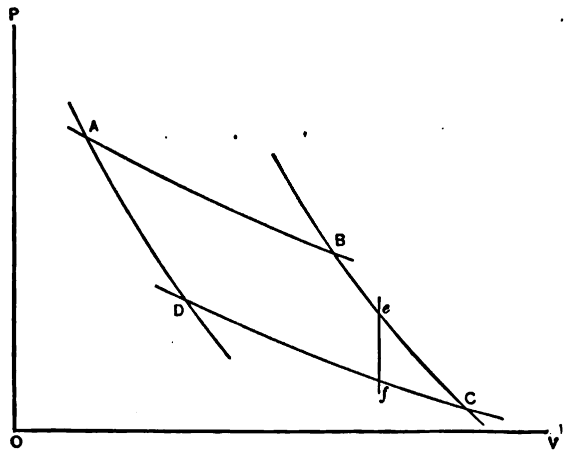


Fig. 2–4. — Pressure Volume Diagram of Isothermal and Adiabatic Changes.

A simple illustration is shown in Figs. 2–4 and 2–5. In Fig. 2–4 is shown a pressure-volume diagram of a working material which expands isothermally from *A* to *B*, then adiabatically from *B* to *C*. It is then compressed isothermally from *C* to *D* and adiabatically from *D* to *A*, when it reaches the initial condition. The mechanical work performed during these operations is proportional to the area of the diagram *A B C D*. This diagram may be

transformed into a temperature entropy diagram very easily, since for that case the isothermal lines $A\ B$ and $B\ C$, would be horizontal and parallel to the line of zero temperature, and the adiabatic lines $A\ D$ and $D\ C$ would be vertical and parallel to the line from which entropy is reckoned. Fig. 2–5 shows the temperature-entropy diagram constructed as described. The area of this diagram shows the heat transferred into work. For the case considered this is a rectangle and its area represents the maximum amount of heat available for work within the temperature limits $A\ B$ and $D\ C$.

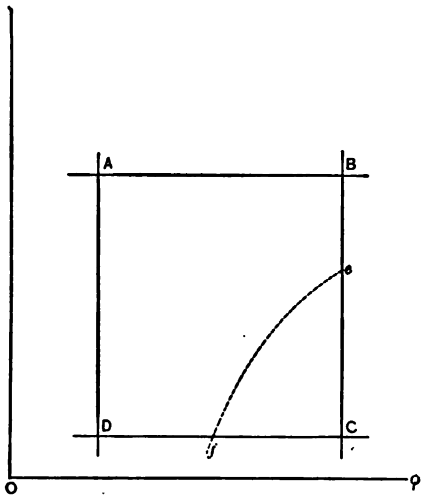


Fig. 2-5. — Entropy-Temperature Diagram.

The case considered is that of the perfect reversible engine which operates in a Carnot cycle, as already described. It appears from the diagram, Fig. 2–5, as well as from the demonstration, that an engine working on this principle can transform the maximum amount of heat into work.

An engine working on any other principle, as for instance that shown in the Fig. 2–2 $A\ B\ D\ E\ F\ G\ A$, will transform a less amount of heat into work. For this case there is both adiabatic expansion $B\ D$ and adiabatic compression $F\ G$. The entropy-temperature diagram for this case is represented, but not to scale, in Fig. 2–6 by the diagram $A\ B\ D\ E\ F\ G$, which by inspection is smaller and shows less heat available for work than the diagram $m\ B\ n\ F$, which is drawn between the same temperature limits.

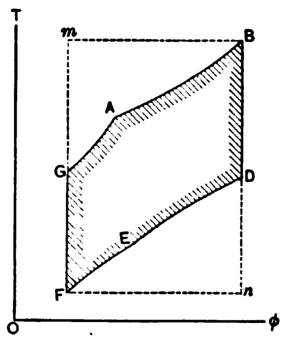


Fig. 2-6.

For the figure $A\ B\ C\ E\ F\ H$, shown in Fig. 2–2, as a pressure-

volume diagram we have isothermal expansion *B C* and isothermal compression *F H*. The entropy-temperature diagram for this case is represented by the diagram, not drawn to scale, in Fig. 2-7, *A B C E F H*, which by inspection is smaller and shows less heat available for work than the diagram *m C n H*, which is drawn between the same temperature limit. The transformation of the pressure-volume diagram as taken on the indicator into entropy-temperature diagram is given at length later in the book.

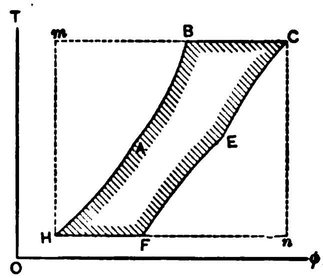


FIG. 2-7.

10. Comparison of Theoretical and Actual Heat Engines. — When a gas after a series of transformations of pressure-volume and temperature passes through a series of intermediate states and of physical and chemical changes and returns to the same condition, in all respects, which it possessed at the beginning of the transformations, it is said to operate in a *closed cycle.*

It is evident that if a change of composition occurs during the course of the cycle the body may return to its initial condition so far as pressure and volume are concerned without returning to its initial condition in other respects; for example, a mixture composed of hydrogen, carbon monoxide, methane, carbon dioxide and nitrogen, with which the cylinder is charged in the initial condition before combustion, may be changed during combustion to the vapor of water, carbon dioxide, and nitrogen, which change would not be shown on the indicator diagram.

In the actual operation of the internal combustion engine the gas or vapor is subjected to periodical changes, as outlined above which do not constitute rigorously closed cycles. The operation is, however, approximately a closed cycle since the state of the working fluid after each series of changes returns to its initial state, from which all the operations as to chemical and physical changes are reproduced as in the preceding phase.

The machine in which the changes take place will be a perfect heat engine if all of the heat disappearing has been transformed into work; it has been demonstrated practically, however, that it

is not possible to utilize all of the heat, consequently it is necessary to expend in heat a larger quantity than its equivalent in units of work. It has already been shown in effect that it is not only necessary to have a place of combustion which produces a high temperature at the beginning of the period of movement, but it is also necessary to have a point of lower temperature which will act as a refrigerator and permit the flow of heat from a higher to a lower temperature level. The quantity of heat Q supplied by the combustion less the amount q taken up by the refrigerator leaves a difference $Q - q$ which may be utilized. From this statement it would appear that the amount of work possible would be increased either by increasing Q or by diminishing q. The temperature of the discharge heat q must evidently be considerably above that of absolute zero because of the difficulty of obtaining a refrigerant of low temperature and of disposing of the heat which would be discharged under such a condition. Generally in practice the temperature of the discharge heat is considerably above that of the surrounding air, which is much in excess of absolute zero.

The ratio of $Q - q$ to Q measures the perfection of the heat engine. This we will call the cyclic efficiency

$$\frac{Q - q}{Q} = \text{cyclic efficiency}$$

It is of great practical interest to know how to determine the value of this coefficient for any case, but we should at first establish the maximum value that can be obtained.

In this connection the cycle of Carnot which is formed of two *isothermal* lines and two *adiabatic* lines, Figs. 2–4 and 2–5, should receive consideration, since it is the one which gives the maximum work returned for the heat expended.

The Carnot cycle is represented in the pressure-volume diagram Fig. 2-4, in which a mass of fluid in its initial state $p_o v_o$, with temperature T_0 as shown at A, expands in volume isothermally to B, at which point it has pressure and volume $p_1 v_1$. From B it expands to C without gain or loss of heat, following the adiabatic $B\,C$. From C to D there is a discharge of heat into a colder body or refrigerator at constant temperature, during which time the volume is reduced from v_2 to v_3. From D to A compression takes

place without gain or loss of heat, which raises the temperature to that of the initial state at A.

In order to carry out the cycle of operation described, sufficient heat must be supplied during the isothermal expansion from A to B to keep the temperature constant; this amount by equation (19) will be $Q = A\ R\ T \log \frac{v_1}{v_0}$. In the adiabatic expansion from B to C, during which there is neither gain nor loss of heat, the relations of the volumes to the temperatures are expressed by equation (16)

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{\gamma - 1}$$

During the third period from C to D heat is discharged at constant temperature and can be expressed as before $q = A\ R\ T \log \frac{v_2}{v_3}$. During the fourth period the quantity of heat remains constant and we have

$$\frac{T_0}{T_3} = \left(\frac{v_3}{v_0}\right)^{\gamma - 1}$$

The cycle as above described is a closed one, and as the heat received and discharged is of constant temperature it follows that

$$\frac{Q}{q} = \frac{T_1}{T_2}$$

The work produced is equal to the area $A\ B\ C\ D$. The efficiency of the cycle is

$$\frac{Q - q}{Q} = \frac{T_1 - T_2}{T_1}$$

This is equal to the ratio of the fall of temperature to the absolute temperature of the initial condition.

The Carnot cycle may also be represented, as already shown, by the temperature-entropy diagram, in which case the diagram will be a rectangle, Fig. 2–5.

It is doubtless true that no better method exists for utilizing the heat furnished by combustion than that of supplying it at constant temperature, permitting the body to expand without gain or loss of heat, discharging it to the refrigerator at a constant temperature, which should be as low as possible, and compressing without gain or loss of heat to the original temperature.

Theoretically it is possible to equal but not to surpass the return from the Carnot cycle. The maximum effect that can be

obtained from a heat engine working between the temperatures T and T', and in which Q is the heat expended, is expressed by the equation

$$Q\left(\frac{T-T'}{T}\right)$$

In order to judge the theoretical value of a cycle on which any heat engine operates it is desirable to calculate at first the coefficient of economy of the proposed cycle, then compare this coefficient with the Carnot coefficient between the same temperature limits.

The knowledge which is given by comparing the cycle of the engine with the Carnot cycle is not sufficient to determine the practical value of the engine, since the effect of friction, shocks due to inertia, and the passive resistance of the various mechanical parts consume a portion of the work supplied by the transformation of heat and do not appear in the useful work delivered by the machine. It is quite possible that machines which have a high degree of perfection for the transformation of heat will still give small return as practical, useful machines.

The hot-air engines for example, which have a perfect cycle of operation, have proved in practice of little value because of the small amount of heat that would pass through a metallic wall in a given time, and as a consequence the return in useful work is small in proportion to the expense of construction. There are few engines of any kind which operate in a cycle approximating that of Carnot, among these should be mentioned the Sterling air engine, which theoretically operates on the Carnot cycle.

The cycle of operation of the steam engine is incomplete in many respects; it however resembles that of Carnot in that heat is received into the engine cylinder, until the valve closes connection with the boiler, at practically constant temperature. After cut-off the steam is expanded approximately without gain or loss of heat. It is then discharged through the condenser at constant temperature. The adiabatic compression is sometimes considered as being performed by the feed pump which supplies water to the boiler. The steam engine cycle is thus seen, when the boiler furnace and boiler feed pump are included as a part, to approximate that of the Carnot cycle.

The Diesel motor is the only gas engine which approximates in the theory of its operation to the Carnot cycle.

CHAPTER III

THEORETICAL COMPARISON OF VARIOUS TYPES OF INTERNAL COMBUSTION ENGINES

1. Throughout the following discussion of the theoretical cycles it will be assumed that the specific heats at constant volume C_v, and at constant pressure C_p, do not vary either with pressure or temperature. It has been shown that they vary, but the question is unsettled. If the variation is such as determined by the experiments of Mallard and Le Chatelier, which are extensively quoted, our present-day gas engine does not admit thermally of any further improvement. In view of this unsettled condition it is best to assume the specific heats constant. It is further assumed, in this theoretical discussion, that the value of $\gamma = \frac{C_p}{C_v}$ is the same for the burned gases as for the fresh fuel mixture.

It is further understood that, wherever heat supplied to a cycle is mentioned, it refers to the lower heating value of the fuel concerned, either per pound or per standard cubic foot, as stated.

2. The cycle receiving heat at constant volume, Beau de Rochas or Otto cycle.

The principles upon which the present-day constant volume combustion engine is based, and which helped it to its commercial success, were first clearly enunciated by Beau de Rochas in a written pamphlet in 1862. It remained for Otto, however, to construct the first practically successful machine operating with this cycle, hence the cycle receiving heat at constant volume is more often known as the Otto cycle.

In what follows let

Q = quantity of heat received by the theoretical cycle,

q = quantity of heat rejected by the theoretical cycle,

then E_c = the cycle efficiency $= \frac{Q-q}{Q}$.

It shows the highest efficiency which an engine could possibly show if it could follow exactly the lines of the theoretical cycle. Practically this can never be realized, but the conditions which determine why the actual thermal efficiency of an engine is always less than the cyclic efficiency will be treated in detail later on.

In Fig. 3–1, at the end of the charging stroke, whether that be in the two- or in the four-cycle, the charge is under a pressure, temperature, and volume determined by the point 1. Adiabatic compression then takes place to 2. A quantity of heat, Q, is next received at constant volume to 3. From 3 to 4 adiabatic expansion takes place, and finally the quantity of heat, q, is rejected along line 4–1 at constant volume.

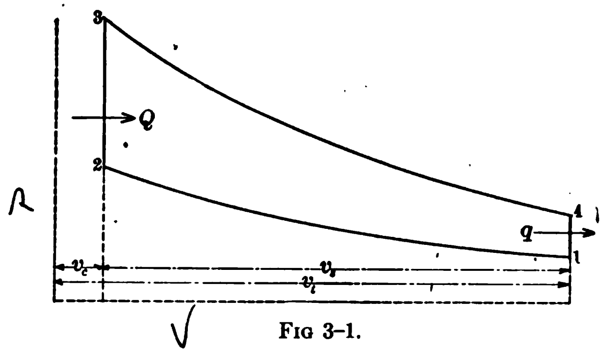


FIG 3–1.

Let the total charge weight be G lbs. This consists of G_a lbs. air, G_g lbs. gaseous fuel, and G_r lbs. burned gases from the previous cycle.

Then

$$Q = GC_v\,(T_3 - T_2) \tag{1}$$

and

$$q = GC_v\,(T_4 - T_1) \tag{2}$$

Hence the heat utilized by the cycle is

$$Q - q = GC_v\,(T_3 - T_2 - T_4 + T_1) \tag{3}$$

and the cyclic efficiency

$$E_c = \frac{Q-q}{Q} = \frac{GC_v(T_3 - T_2 - T_4 + T_1)}{GC_v(T_3 - T_2)} \tag{4}$$

$$= \frac{T_3\left(1 - \frac{T_4}{T_3}\right) - T_2\left(1 - \frac{T_1}{T_2}\right)}{T_3 - T_2} \tag{5}$$

Again, in Fig. 3–1, let v_c = clearance volume, and v_s = stroke volume, so that $v_t = v_c + v_s$ = total volume. We may write, since compression and expansion are assumed adiabatic,

$$p_2 v_c^\gamma = p_1 v_t^\gamma \tag{6}$$

and

$$p_3 v_c^\gamma = p_4 v_t^\gamma. \tag{7}$$

Dividing (6) by (7), $\frac{p_2}{p_3} = \frac{p_1}{p_4}$. (8)

Now $\frac{T_2}{T_3} = \frac{p_2}{p_3}$ and $\frac{T_1}{T_4} = \frac{p_1}{p_4}$

hence finally $\frac{T_2}{T_3} = \frac{T_1}{T_4}$ or $\frac{T_4}{T_3} = \frac{T_1}{T_2}$ (9)

With the aid of (9), equation (5) may be written

$$E_c = \frac{T_3\left(1 - \frac{T_1}{T_2}\right) - T_2\left(1 - \frac{T_1}{T_2}\right)}{T_3 - T_2}$$

$$= 1 - \frac{T_1}{T_2} \tag{10}$$

There are two other ways of stating the cyclic efficiency which will be developed next.

Again we can write

$$p_1 v_t^\gamma = p_2 v_c^\gamma \tag{11}$$

also

$$\frac{p_1 v_t}{T_1} = \frac{p_2 v_c}{T_2} \tag{12}$$

Dividing (11) by (12)

$$T_1 v_t^{\gamma - 1} = T_2 v_c^{\gamma - 1}$$

or

$$\frac{T_1}{T_2} = \left(\frac{v_c}{v_t}\right)^{\gamma - 1} = \frac{1}{r^{\gamma - 1}} \tag{13}$$

in which r = the compression ratio $\frac{v_t}{v_c}$.

Hence

$$E_c = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{r^{\gamma - 1}} \tag{14}$$

Finally, raise equation (12) to the γ power

$$\frac{p_1^{\gamma} v_t^{\gamma}}{T_1^{\gamma}} = \frac{p_2^{\gamma} v_c^{\gamma}}{T_2^{\gamma}} \tag{15}$$

and divide (11) by (15). We shall have

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}, \tag{16}$$

from which

$$E_c = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} \tag{17}$$

From an examination of equations (14) and (17) it will be seen that E_c depends upon r or p_2, and $\gamma = \frac{C_p}{C_v}$.

To make clear the influence of the value of γ, assume that in two given cases the value of $r = 5$, but that in the first case a rich gas mixture with $\gamma = 1.35$, in the second case a lean gas mixture with $\gamma = 1.39$ be employed. Then for the two cases we shall have

$$E_c = 1 - \frac{1}{(5)^{.35}} = .431$$

and

$$E_c = 1 - \frac{1}{(5)^{.39}} = .466$$

The advantage in favor of the lean gas mixture is therefore

$$\frac{.466 - .431}{.431} = 8.1\ \%$$

Besides this the use of lean gas mixtures in practice usually shows a smaller jacket water loss, owing to the lower mean temperature of the entire cycle.

The value of E_c also depends upon the ratio $\frac{p_1}{p_2}$, according to eq. (17). The smaller this ratio the greater the efficiency. But the value of p_1, the suction pressure, is almost entirely out of our control, so that the problem narrows down to making p_2, the compression pressure, as high as possible. This brings out clearly the value of high compression. The practical limits to this statement will be pointed out later on.

To show the combined influence upon E_c of r or p_2 and γ the following table, from Güldner, is given:

Cyclic Efficiencies, E_c., for the Otto Cycle

$r =$	2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0
$\gamma = 1.20$	.129	.167	.197	.221	.242	.260	.275	.301	.322	.340	.356	.367
$= 1.25$	.159	.205	.240	.269	.290	.313	.331	.361	.385	.405	.423	.438
1.30	.188	.241	.281	.313	.343	.363	.383	.416	.442	.464	.483	.499
1.35	.216	.274	.319	.355	.384	.409	.431	.466	.494	.517	.537	.553
1.40	.248	.313	.363	.402	.434	.460	.483	.520	.550	.574	.594	.611

3. The cycle receiving heat at a constant pressure, the Brayton cycle, and the approximate Diesel cycle of to-day.

The Brayton Cycle.

In the older machines of the Brayton type, suction and compression of the fuel mixture or of air took place in one cylinder,

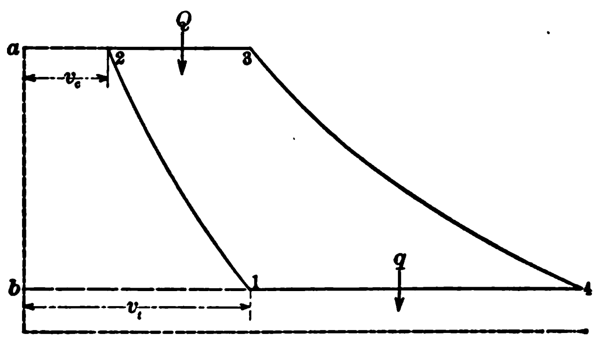


Fig. 3-2.

while the combustion, expansion, and exhaust took place in another. In Fig. 3-2, area b 1 2 a b represents the pump diagram, area a 3 4 b a the diagram from the power cylinder. For the purpose of theoretical discussion of this cycle, the two diagrams may be combined as shown, giving in area 1 2 3 4 the useful work developed.

The compression line, 1–2, and the expansion line, 3–4, are again assumed adiabatic. Expansion is carried to exhaust pressure. The quantity of heat Q is received along 2–3, the amount q is rejected along 4–1, at constant pressure in both cases.

Let the charge weight be G pounds made up as in the previous case. Then

$$Q = GC_p\,(T_3 - T_2) \text{ B. T. U.}$$

and

$$q = GC_p\,(T_4 - T_1) \text{ B. T. U.}$$

Hence
$$E_c = \frac{Q-q}{Q} = \frac{GC_p(T_3-T_2-T_4+T_1)}{GC_p(T_3-T_2)}$$

$$= 1 - \frac{T_4-T_1}{T_3-T_2} \tag{18}$$

But from the adiabatic law we may write

$$p_1 v_1{}^{\gamma} = p_2 v_2{}^{\gamma}. \tag{19}$$

and

$$p_4 v_4{}^{\gamma} = p_3 v_3{}^{\gamma}. \tag{20}$$

Divide (19) by (20).

$$\left(\frac{v_1}{v_4}\right)^{\gamma} = \left(\frac{v_2}{v_3}\right)^{\gamma} \text{ or } \frac{v_1}{v_4} = \frac{v_2}{v_3} \tag{21}$$

Also,

$$\frac{p_1 v_1}{T_1} = \frac{p_4 v_4}{T_4}\text{, or since } p_1 = p_4,\ \frac{v_1}{T_1} = \frac{v_4}{T_4} \tag{22}$$

and similarly

$$\frac{v_2}{T_2} = \frac{v_3}{T_3} \tag{23}$$

From (22)

$$\frac{v_1}{v_4} = \frac{T_1}{T_4} \tag{24}$$

From (23)

$$\frac{v_2}{v_3} = \frac{T_2}{T_3} \tag{25}$$

Equations (25) and (24) in combination with (21) finally give

$$\frac{T_1}{T_4} = \frac{T_2}{T_3} \text{ or } \frac{T_1}{T_2} = \frac{T_4}{T_3} \tag{26}$$

Substituting (26) in (18), we have

$$E_c = 1 - \frac{T_1}{T_2} = 1 - \frac{T_4}{T_3} \tag{27}$$

But it has already been shown, equation (13), that for adiabatic compression

$$\frac{T_1}{T_2}=\left(\frac{v_c}{v_t}\right)^{\gamma-1}=\frac{1}{r^{\gamma-1}}$$

so that finally the cyclic efficiency for combustion at constant pressure is

$$E_c=1-\frac{1}{r^{\gamma-1}} \qquad (28)$$

which is the same as for combustion at constant volume.

The Diesel Cycle of to-day.

The Diesel cycle of to-day approximates the constant-pressure form outlined in Fig. 3–3. Compression line 1–2 and expansion

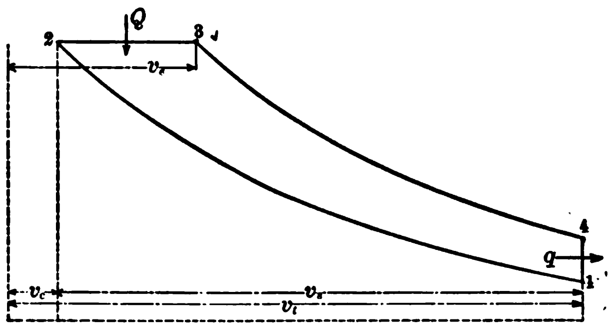


FIG. 3–3.

line 3–4 are assumed adiabatic. Heat is received at constant pressure along line 2–3, and rejected at constant volume along line 4–1. In the Otto cycle, with the machine at full load, the ratio of compression is equal to the ratio of expansion. In the Diesel, the ratio of compression $\frac{v_t}{v_c}$, (see Fig. 3–3), is always greater than the ration of expansion, $\frac{v_t}{v_e}$.

Let the charge weight again be G pounds. Diesel machines, as constructed, are oil engines, so that the increase of charge weight along line 2–3 is small and may be neglected without serious error. That is, we may assume G constant for the cycle. If the investigation, however, is carried through for a gas, especially a lean gas, this assumption is not permissible.

To develop the efficiency formula, we may write

$$Q = GC_p\,(T_3 - T_2). \tag{29}$$

$$q = GC_v\,(T_4 - T_1). \tag{30}$$

Now from Fig. 3–3

$$\frac{v_c}{T_2} = \frac{v_e}{T_3} \text{ or } T_3 = T_2\frac{v_e}{v_c} = T_2\,\delta \tag{31}$$

where, δ = ratio of cut-off volume to clearance volume.

Also

$$\frac{p_4}{T_4} = \frac{p_1}{T_1} \text{ from which } T_4 = T_1\frac{p_4}{p_1} \tag{32}$$

But from the adiabatic law

$$p_4 v_4^{\gamma} = p_3 v_3^{\gamma} \text{ from which } p_4 = \frac{p_3 v_3^{\gamma}}{v_4^{\gamma}}$$

and

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma} \text{ from which } p_1 = \frac{p_2 v_2^{\gamma}}{v_1^{\gamma}}$$

Hence (32) may be written

$$T_4 = T_1\frac{\dfrac{p_3 v_3^{\gamma}}{v_4^{\gamma}}}{\dfrac{p_2 v_2^{\gamma}}{v_1^{\gamma}}} = T_1\left(\frac{v_3}{v_2}\right)^{\gamma} = T_1\left(\frac{v_e}{v_c}\right)^{\gamma} = T_1\delta^{\gamma} \tag{33}$$

Substituting (31) and (33) in (29) and (30) respectively,

$$Q = GC_pT_2\,(\delta - 1).$$

$$q = GC_vT_1\,(\delta^{\gamma} - 1).$$

The cyclic efficiency for the Diesel cycle consequently is

$$E_c = \frac{Q - q}{Q} = 1 - \frac{q}{Q} = 1 - \frac{GC_vT_1(\delta^{\gamma} - 1)}{GC_pT_2(\delta - 1)}$$

$$= 1 - \frac{T_1}{T_2}\,\frac{(\delta^{\gamma} - 1)}{(\delta - 1)\gamma}$$

But it has shown, equation (13), that $\dfrac{T_1}{T_2} = \dfrac{1}{r^{\gamma-1}}$

hence finally

$$E_c = 1 - \frac{1}{r^{\gamma-1}} \frac{(\delta^{\gamma} - 1)}{\gamma(\delta - 1)} \tag{34}$$

Equation (34) shows that the expression for the theoretical efficiency of the Diesel cycle is the same as that for the Otto and the Brayton, with the exception of the factor $\frac{\delta^{\gamma} - 1}{\gamma(\delta - 1)}$. The efficiency thus depends not only upon r and γ, but also upon δ, that is, finally, on the volume at cut-off.

In actual practice the cut-off volume v_e is about 10 per cent of the stroke at full load. With a compression ratio of $r = 13$, this makes δ about $= 2.5$. To show the influence of the factor δ upon E_c, assume $\gamma = 1.35$. The efficiencies for full load, $\delta = 2.5$, an overload, $\delta = 3.0$, and some partial load, $\delta = 1.5$, will be as follows:

Cyclic Efficiency E_c for the Diesel cycle.

For $\delta = \frac{v_e}{v_c} =$	1.5	2.5	3.0
and for $r = 13$, and $\gamma = 1.35$, $E_c =$	.560	.509	.487.

It appears from this that, other conditions remaining the same, the smaller the value of δ, the greater E_c. This result is actually borne out in practice, within limits, where a large number of tests of Diesel engines have often shown a greater thermal efficiency at three-quarters than at full load. That this condition does not hold for still lower loads is due to other circumstances.

4. Comparison of Various Cycles.

The question of the best gas-engine cycle has often been discussed. In general, there is no best gas-engine cycle, but that cycle should be chosen which will give the best return for the practical conditions existing.

To give some insight into the problem of choosing the cycle best adapted to given conditions, we will first obtain some theoretical basis of comparison, and show afterwards how this is affected in practice.

Many methods of comparison have been employed by various writers, but the following, due originally to E. Meyer * seems to the writer to be the clearest and most comprehensive.

* Zeitschrift des Vereins deutscher Ingenieure, 1897, p. 1108.

Using the Carnot cycle as a basis of comparison, it is clear that the amount of heat transferred into mechanical work depends only upon the total amount of heat and the temperature difference in the cycle. But to become available as a basis for the comparison of gas-engine cycles, only that Carnot cycle can be used for which the heat element δq_1, supplied at a constant temperature T_1, and the heat element δq_2 rejected at a constant temperature T_2, are of infinitesimal amount, so that the two adiabatics forming the rest of the cycle are infinitely close together. Then every closed cycle may, by a number of adiabatics, be divided into an infinite number of elementary cycles, Fig. 3–4,

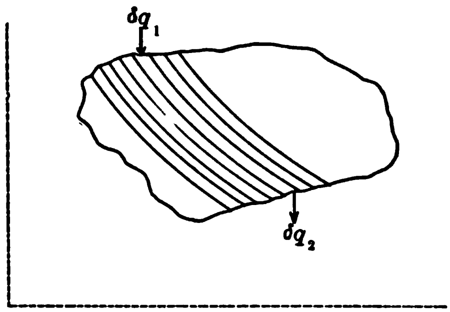


FIG. 3–4.

for each of which we may with very small error assume that the heat element δq_1 is supplied at a constant temperature T_1, and the heat element δq_2 is rejected at the constant temperature T_2. The efficiency of one of these elementary Carnot cycles may be expressed for say the *nth* cycle by

$$E_{c(n)} = \frac{T_{1(n)} - T_{2(n)}}{T_{1(n)}}$$

From the above equation we at once derive the important requirement that for best efficiency each heat element δq_1 should be supplied to the working fluid at the highest temperature possible, and each element δq_2, should be rejected at the lowest possible temperature, that is, the temperature limits should be as wide as possible.

It should be observed, however, that the above requirement

does not mean that the sum total of all the heat elements δq_1 must be supplied isothermally, or that the sum total of the elements δq_2 rejected must be taken up isothermally. For in general in gas engines the various successive heat elements are not available at constant temperature, and in fact in following out the requirement above outlined for best efficiency, one would be led to constantly change the temperature of supply and of rejection to widen the temperature limits as far as possible. It is clear, therefore, that only in the case where all the heat elements supplied are available at constant temperature, and all those rejected can be taken up at constant temperature, as is the case in a steam engine, will the Carnot cycle represent the ideal.

Now consider one of the elementary Carnot cycles, into which the gas-engine cycle has been divided, by itself, Fig. 3-5.

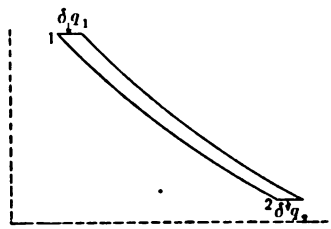


FIG. 3-5.

For the point 1 we may write

$$T_1=\frac{p_1v_1}{R}$$

and since during the supply of the infinitesimal heat element δq, the values of p_1 and v_1 increase only by infinitesimal amounts, we may assume without great error that the volume and pressure at the end of the heat supply are also represented by v_1 and p_1 respectively. Similarly we may write for the point 2

$$T_2=\frac{p_2v_2}{R}$$

The efficiency of the elementary cycle under discussion is then

$$E_c=\frac{T_1-T_2}{T_1}=\frac{p_1v_1-p_2v_2}{p_1v_1}=1-\frac{p_2v_2}{p_1v_1}$$

But since the other enclosing lines are adiabatics, we have

$$p_1 v_1^{\gamma} = p_2 v_2^{\gamma}, \text{ from which}$$

$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}}$$

Finally, therefore,

$$E_c = 1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{\frac{1}{\gamma}} = 1 - \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \tag{35}$$

In connection with the last equation it should be observed that while the expression $E_c = \frac{T_1 - T_2}{T_1}$ is based upon the general laws of thermodynamics, and is therefore applicable to all working fluids without exception, equation (35) has been derived from this by the use of equations $\frac{p\,v}{T} = R$, and $p\,v^{\gamma} =$ constant. Equation (35) is therefore strictly applicable to gas only.

Equation (35) leads to the following important deduction: *To obtain the best efficiency in a closed cycle, it is necessary to supply each heat element at such a pressure p_1 and to reject the part not transformed into work at such a pressure p_2, that the ratio $\frac{p_1}{p_2}$ shall be as large as possible.*

As previously pointed out, however, in the case of the gas engine, the pressure p_2 cannot well be below atmosphere because the use of a vacuum is neither practicable nor economical. Hence the meaning of equation (35) narrows down to the requirement *to introduce each heat element at highest possible pressure p_1, and to reject the part not used at a pressure as close to atmosphere as possible.*

With this knowledge it becomes easy to compare the various gas-engine cycles among themselves to determine which of the pressure lines at which heat is supplied gives the best guarantee of efficiency.

Figure 3–6 represents the cycle with combustion at constant volume, Fig. 3–7 the constant pressure cycle, and Fig. 3–8 the cycle with isothermal combustion. The heat is rejected in each case along a constant-pressure line as nearly as possible to atmosphere. It is assumed also in each case that the pressure and temperature at the end of compression shall be the same, that is, that p_c and T_c in Fig. 3–6 shall be equal to p_c and T_c in Fig. 3–7, etc.

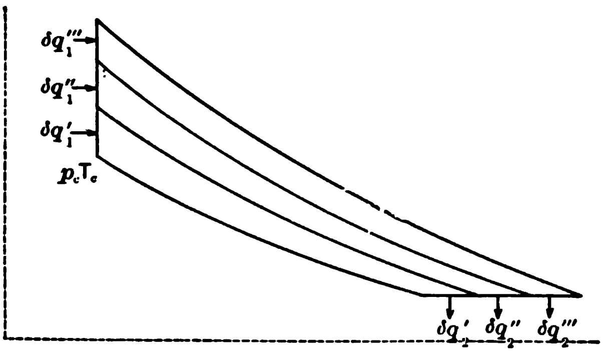


FIG. 3-6.

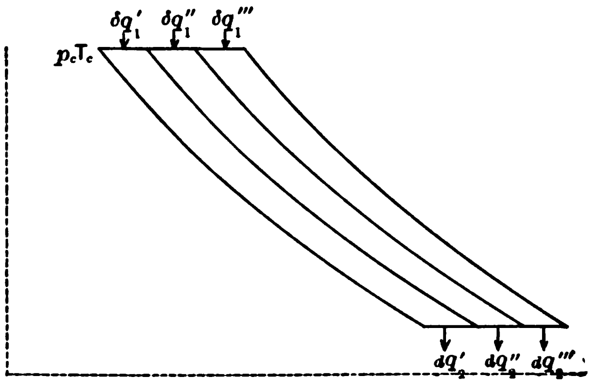


FIG. 3-7.

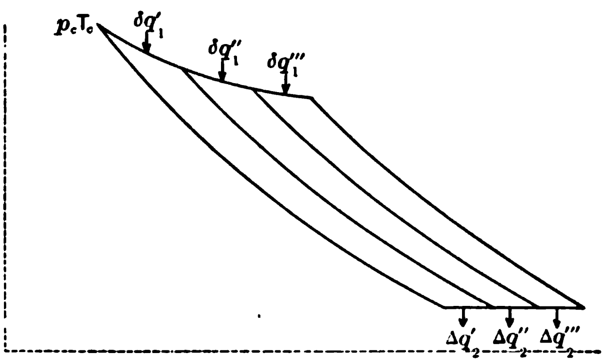


FIG. 3-8.

Now assume that each cycle is divided by adiabatics into an infinite number of elementary cycles. For convenience only three of these are indicated in each figure. With combustion at constant volume Fig. 3-6, each heat element, $\delta q_1'$, $\delta q_1''$, etc., supplied serves to raise the temperature and especially the pressure for the one succeeding it, and since expansion is carried to atmosphere in each case, it is plain that this method of combustion tends of itself to fulfil the requirement above outlined. On account of the increasing pressure ranges, the efficiency of each elementary cycle is in this case greater than that of the one just preceding it.

The same course of reasoning applied to combustion at constant pressure, Fig. 3-7, will show that each heat element, $\delta q_1'$, $\delta q_1''$, etc., while it serves to raise the temperature does not raise the pressure for the next succeeding element. The expansion being again in each case to atmosphere, the pressure ranges for all the elementary cycles, and hence also the efficiency, is the same.

The case of combustion at constant temperature, Fig. 3-8, is even less favorable. Here each heat element, $\delta q_1'$, $\delta q_1''$, etc., does not raise the temperature for the succeeding element, but is accompanied by a decrease of pressure. The exhaust pressure being unchanged, this means a smaller pressure range for each succeeding elementary cycle, and consequently a steadily decreasing efficiency.

Looking next at the heat discharged from each elementary cycle, we may write:

$$\delta q_2' = dq_2' = \Delta q_2'$$
$$\delta q_2' > \delta q_2'' > \delta q_2'''$$
$$dq_2' = dq_2'' = dq_2'''$$
$$\Delta q_2' < \Delta q_2'' < \Delta q_2'''$$

From all this it follows that, starting with the same pressure of compression, the constant-volume combustion is more efficient than that at constant pressure; and this in turn is more efficient than isothermal combustion.

Isothermal combustion, being so obviously inferior to the other two in theory, is also difficult to carry into operation practically, and for these reasons is practically obsolete.

5. Practical conditions affecting the choice of best cycle for any given case.

It is evident from the preceding article that the most important item in the efficiency question is the pressure at the end of compression. The aim should be to use as high a pressure as possible in order to introduce the first heat element at the highest possible efficiency. But the compression pressure governs the maximum pressure and temperature occurring in the cycle, and that brings us to a consideration of the pressure and temperature limits.

Of these two the temperature limit plays but a secondary part, because it is possible to operate on a cycle whose maximum temperature may be 3000 degrees Fahrenheit for the reason that

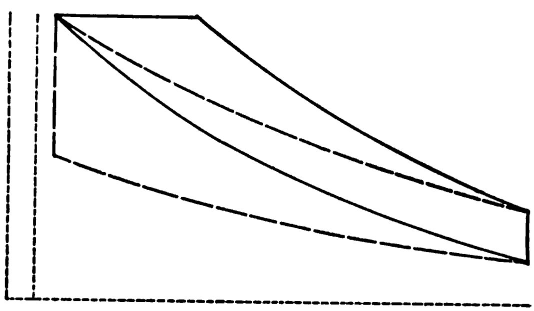


Fig. 3-9.

these maximum temperatures exist but for a very short time. The pressure limit is more important, because no matter how short a time the maximum pressure lasts, the driving mechanism of the machine must be built for this pressure. In modern practice 550-600 pounds seems to be about the maximum pressure limit that can be economically handled. Herein we find a condition which may modify the conclusion arrived at in the previous article as regards comparative efficiency of combustion at constant volume and at constant pressure.

Assuming the more practical condition, that the maximum pressures in the two cycles, instead of the compression pressures, shall be the same, the diagrams would be placed as shown in Fig. 3-9, in which the broken line represents the Otto cycle. It is evident in such a case that only the last heat element of the

Otto cycle is introduced at the same efficiency as the first, and consequently all, of the heat elements of the cycle at constant volume. Hence, upon these premises, the constant-pressure cycle is more efficient than the constant volume cycle. Under some conditions, however, favorable to the Otto cycle, this advantage of the constant-pressure cycle is not very great. Thus in a Diesel engine cutting off at full load at 10 per cent, $\delta = 2.5$, assume $r = 13$, and $\gamma = 1.41$, then E_c will be .564. An Otto engine cycle of the same maximum pressure limit, = about 460 pounds, would show a compression pressure of from 190–225 pounds, r would be about equal to 7, and with $\gamma = 1.41$, E_c would be .550. This shows a gain for the Diesel engine of only 1.5 per cent, but it should be pointed out that a compression pressure of 200 pounds in an Otto cycle can only be reached with extremely lean fuel gases, or with separate fuel and air compression. However that may be, the advantage of the constant-pressure cycle over the constant-volume cycle, presupposing equal maximum pressures in each, is comparatively small, and hence the undoubted gain that the Diesel engine shows in practice over the average constant-volume engine is by some writers attributed not so much to the cycle as to the greater perfection of combustion. That this is approximately true has been proven by Güldner who constructed engines operating on the Otto cycle upon the most advanced ideas, and obtained efficiencies fully as good as those obtained by Diesel.

A second limit set to the compression pressure that can be carried is due to pre-ignition. All fuel mixtures will ignite spontaneously if the temperature becomes high enough, but the critical temperature varies greatly for the different fuels. This fact directly governs the compression pressure. While it is hardly advisable to use more than say 80–90 pounds in the case of a gasoline mixture, a blast furnace gas mixture will easily stand 150 pounds without pre-ignition. Since the efficiency of the cycle depends directly upon the compression pressure, as above shown, we should expect a better efficiency for blast furnace gas than for gasoline, and this is actually so in practice. The difference, however, is due to the nature of the fuel. A remedy for this state of affairs would be to compress the air separately and introduce the gasoline or other fuel oil only at the moment

combustion is desired. This leads to the constant-pressure cycle, and thus the fuel to be employed should be considered in the choice of cycle.

A final point to be considered is this: By theory, the greater the compression pressure, the higher the efficiency. This holds for either type of cycle. But how far can this compression be carried, outside of the questions of upper pressure limit and pre-ignition above considered, before the added gain due to higher compression is in practice balanced or overbalanced by attendant losses.

The following discussion of this question by Güldner, as applied to combustion at constant volume, is instructive.

The efficiency of any engine should not be judged upon cylinder performance, but upon the performance at the shaft. Usually this is called the "thermal efficiency per brake horse-power," but a shorter and more expressive term which the authors prefer and will use in this treatise, is "Economic Efficiency E_e."

Now

$$E_e = E_t E_m = X E_c E_m.$$

where

E_t = thermal efficiency per indicated horse-power.

E_m = mechanical efficiency of engine $= \dfrac{\text{Brake H. P.}}{\text{Indicated H. P.}}$

E_c = cyclic efficiency as derived in the preceding articles =

$$= 1 - \frac{1}{r^{\gamma-1}} \text{ for combustion at constant volume.}$$

and

X = a factor such that $E_t = XE_c$, so that X is always less than 1.

The mechanical efficiency may be expressed by

$$E_m = \frac{p_i - p_f}{p_i}$$

where

p_i = mean indicated pressure per square inch of piston,

and

p_f = pressure lost in friction per square inch of piston.

Hence

$$E_e = XE_c \frac{p_i - p_f}{p_i} = X\left(1 - \frac{1}{r^{\gamma-1}}\right)\frac{p_i - p_f}{p_i}$$

The value of E_e therefore depends upon the relation between the factors X, p_i, and p_f, as modified by a variation in r, that is, in the pressure of compression. The change in the value of X due to a variation in r is quite unknown, and is in any case so small that it may be neglected. Regarding the relation between p_i, p_f, and r, numerous tests have shown that, as r increases, the mechanical efficiency $\frac{p_i - p_f}{p_i}$, is at first quite constant, but that after a certain point it commences to decrease quite rapidly. This is due to the fact that for the first part of the range any increase in p_f, due to an increase in r, is counterbalanced by a corresponding gain in p_i. Beyond a certain point, however, there is a loss in p_i, owing to the fact that the fuel mixture has to be made more and more lean to prevent pre-ignition, and the mechanical efficiency consequently decreases. *Hence we have the net result that as long as an increase in p_f, due to an increase in r, is met by a proportionate gain in p_i, the Economic Efficiency E_e will increase. Just as soon, however, as, with increase in r, the gain in p_i is no longer sufficient to overcome the increase in p_f, E_e will commence to decrease, and the economic compression limit will have been passed.*

Tests and computations have shown that up to $r = 6$, which means a compression pressure of about 160 pounds, the mechanical efficiency does not change materially, but that beyond this it commences to decrease. This is probably due to the necessary increase in the size of machine parts due to the greater pressures and to the fact that leaner mixtures than would ordinarily be used must be employed to prevent pre-ignition, with a corresponding loss in the value of p_i. Beyond $r = 6$ the gain in E_e is small and at $r = 10$ it practically ceases to increase. *Hence we conclude that for combustion at constant volume the use of values of r greater than about 8, for which the compression pressure equals about 225 pounds, is no longer accompanied by any useful gain in the economic efficiency of the machine.* This statement does not apply to combustion at constant pressure because its E_c follows

a different law, and E_m decreases more slowly on account of the generally greater value of p_i.

Finally, in laying out an Otto cycle, to obtain high engine capacity per unit volume of cylinder means making the maximum pressure as high as possible to obtain a high value of p_i. With fuel mixtures which cannot stand a high amount of compression this would mean the use of rich gas mixtures. Where the compression can be made higher, the ratio of $\frac{\text{max. pressure}}{\text{comp. pressure}}$, instead of being from 4–5, may be profitably made from 2.5 to 3 by the use of leaner mixtures, so that the maximum pressure shall be in the neighborhood of say 450 pounds. The chances are, as the gas engine develops, that higher maximum pressures will be employed, but according to Güldner the use of working pressures exceeding 600 pounds is neither economical nor safe.

CHAPTER IV

THE VARIOUS EVENTS OF THE CONSTANT VOLUME AND THE CONSTANT-PRESSURE CYCLES AS MODIFIED BY PRACTICAL CONDITIONS

1. In the previous chapter the various cycles were discussed and compared on theoretical grounds. For this purpose several things were assumed which in practice are only approximately true; thus compression and expansion lines were assumed adiabatic and the surrounding walls impermeable to heat. It is, however, true that the heat interchange between the charge and the walls may be such as to give a line on a diagram which strictly follows the adiabatic law. Such a line is by some writers called a false or pseudo-adiabatic. Again it has been assumed in the theoretical discussion that ignition is perfect, that the composition of the charge is uniform, and that combustion is complete and perfect; none of these things quite obtain in practice and all the variations have their influence upon engine performance. Thus it happens that the cyclic efficiency above computed is in any given case never realized, but that the actual thermal efficiency is always less than E_c.

The following paragraphs will point out these modifications more in detail.

2. **The four-stroke Otto cycle.**

(*a*) The Suction Stroke. At the end of the exhaust stroke, the clearance volume V_c, Fig. 4-1, is filled with burned gases under a pressure p_e and a temperature T_e. The weight of these gases can only be approximately computed, since nothing definite is known of the temperature T_e. It is in most cases probably between 12-1400 degrees Fahrenheit, while the pressure p_e in well-designed machines may be from 16-18 pounds absolute, but circumstances may alter these figures considerably. At the commencement of the suction stroke the pressure falls from p_e to the suction pressure p_s along a curve determined by the re-expansion of the burned gases in the clearance spaces. Only after this re-expansion will the fresh charge be drawn into the cylinder. It is thus seen that the

volumetric efficiency, E_v, of the cylinder, that is, the ratio $\frac{\text{volume of fresh mixture}}{\text{volume of piston displacement}}$, depends directly upon the weight of burned gases remaining, thus affecting cylinder capacity. If through bad form of combustion chamber, too small an exhaust opening, or a restricted exhaust pipe, the exhaust pressure should be kept too high, or too much burned gas remain behind, this effect of re-expansion will be more marked than above stated. A too early closure of the exhaust valve may have the effect shown in Fig. 4–1 in dotted line. It is evident, however, that in the ordinary four-cycle engine without scavenging this loss of volumetric efficiency will always be present, depending upon the clearance volume.

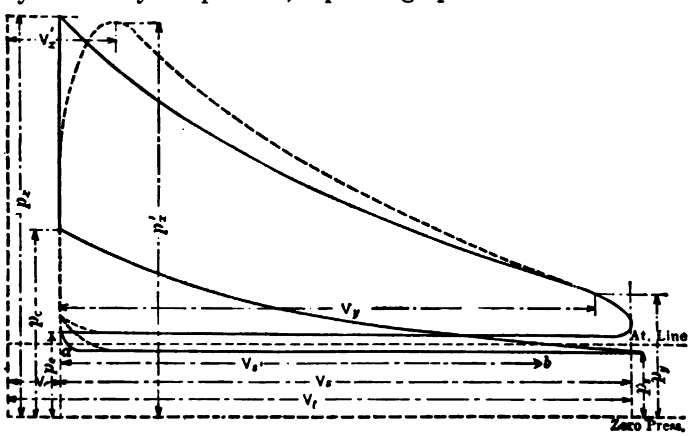


FIG. 4–1.

A second factor affecting the volumetric efficiency of the cylinder is the suction pressure p_s. At the end of the suction stroke the cylinder contains a volume of gas, v_t, made up partly of burned gas and partly of fresh mixture, under a pressure p_s. The compression stroke raises this amount of gas to the pressure p_c with a volume v_c. The compression curve crosses the atmospheric line when the stroke volume is only $v_{s'}$ and not the full volume v_s. Hence the volume $v_s - v_{s'}$ represents a loss in volumetric efficiency, and this loss is the greater the smaller p_s. It follows that the inlet pipes and valves should be so designed as to cause a minimum suction "under-pressure," that is, to keep p_s as close to atmospheric pressure as possible. This also makes clear why vaporizers and carbureters always decrease engine capacity somewhat.

Thus there is a loss of volumetric efficiency, and hence of engine capacity, at each end of the suction line. The real volumetric efficiency, E_v, is found in any given case by dividing the volume represented by the line *a–b*, measured along the atmospheric line, by the volume v_s of the piston displacement.

Attention should at this point be called to the fact that certain systems of speed regulations depend upon the variation in the suction pressure. By throttling the mixture from the beginning of the stroke, or by cutting off the supply completely at a given point in the stroke, p_s is increased or decreased depending on the load on the engine, thus directly controlling the charge volume, and hence the engine capacity. This is explained more in detail in the chapter on governing.

Since it is quite evident that neither the loss by re-expansion or that due to the suction pressure at the end of the suction stroke can be entirely avoided, it becomes interesting, at least from the point of design, to know approximately what values of E_v to expect in different types of engines. Naturally E_v decreases as engine speed increases, because since high-speed engines are usually small engines, the difficulty of placing valve openings large enough to prevent serious throttling of the charge becomes greater as speeds increase. Computations are of little avail in this matter since little is definitely known of the temperature of the charge at the end of the suction stroke. For that reason more reliance is to be placed in figures based upon practical experience. The following table is due to Güldner:

	E_v	p_s lbs. per sq. inch absolute
1. Slow-speed engines with mechanically operated inlet valve	.88 – .93	12.9 – 13.7
2. Slow-speed engines with automatic inlet valve	.80 – .87	12.5 – 13.2
3. High-speed engines with mechanically operated inlet valve	.78 – .85	11.7 – 12.5
4. High-speed engines with automatic inlet valve	.65 – .75	11.4 – 12.2
5. Very high-speed engines with automatic inlet valves and air cooling	.50 – .65	8.8 – 11.0

Suction gas generators and vaporizers may in unfavorable cases decrease the above figures for E_v as much as .05.

(*b*) The Compression Stroke. — The compression line may

be taken to follow the general law pv^n = constant. During the first part of the stroke there is probably a flow of heat from the walls to the comparatively cool charge, but this is soon overbalanced by the heat of compression, so that during the last and greatest part of the stroke the flow is into the walls. The compression curve is therefore rarely an adiabatic pv^γ = constant, where $\gamma = \frac{C_p}{C_v}$, as computed for the charge. In most cases the line is intermediate between an adiabatic and an isothermal, and strictly also the value of the exponent, n, is not constant along the entire line. The value of n in actual cases lies between 1.30 and 1.38, with an average of about 1.35. In cases of very ineffective cooling n may exceed $\gamma = \frac{C_p}{C_v}$. It should also be noted that leaky pistons and valves cause a flattening of the compression curve, which apparently decreases the true value of n.

From the equation pv^n = constant, we may derive the following equation for the absolute pressure p_c at the end of compression, see Fig. 1.

$$p_c = p_s\left(\frac{v_t}{v_c}\right)^n = p_s r^n$$

The absolute temperature at the pressure p_c will be

$$T_c = T_s\left(\frac{p_c}{p_s}\right)^{\frac{n-1}{n}} = T_s r^{\frac{n-1}{n}}$$

This equation requires an assumption for the value of T_s, the temperature at the end of the suction stroke. As already stated, not a great deal is known about this. S. A. Moss states that experiments have shown it to be between 200–300 degrees Fahrenheit, that is, 660–760 degrees absolute, but gives no details. Schöttler in his examples on type-cycles assumes in most cases 350 degrees Centigrade absolute, = 632 degrees Fahrenheit absolute.

The clearance volume required to produce a pressure p_c and a temperature T_c will be

$$v_c = v_t\left(\frac{p_s}{p_c}\right)^{\frac{1}{n}} = v_t\left(\frac{T_s}{T_c}\right)^{\frac{1}{n-1}}$$

The following table shows values for the absolute compression pressure, p_c and the end temperature T_c for various values of n, of r, the ratio of compression, and of T_s. The value of p_s has been assumed at 12.5 pounds absolute:

Ratio of Compression r =		2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0
n = 1.25	For p_s = 12.5, p_c =	29.7	39.2	49.3	59.9	70.7	81.8	93.2	117.5	142.0	168.0	195.0	222.0
	when T_s = 600, T_c =	690	720	747	770	792	811	828	858	885	910	930	951
	650	748	780	811	835	858	877	896	930	958	983	1010	1028
	700	805	840	875	900	924	945	965	1002	1032	1062	1088	1110
	750	863	900	937	956	990	1013	1036	1073	1107	1137	1163	1188
	800	921	960	1000	1027	1056	1080	1105	1144	1180	1212	1240	1267
n = 1.30	For p_s = 12.5, p_c =	30.7	41.2	52.0	63.7	75.7	83.3	101.2	128.2	156.7	186.5	217.5	250.0
	when T_s = 600, T_c =	703	740	772	800	825	847	868	905	938	966	1000	1018
	650	763	803	837	866	895	917	942	980	1015	1043	1076	1102
	700	821	864	900	933	964	987	1012	1055	1093	1128	1160	1187
	750	880	925	965	1000	1030	1058	1087	1130	1170	1210	1242	1272
	800	937	987	1028	1065	1100	1130	1157	1206	1250	1290	1322	1355
n = 1.35	For p = 12.5, p_c =	31.9	43.0	55.1	67.8	81.2	95.2	109.5	140.5	162.4	206.5	242.2	300
	when T_s = 600, T_c =	718	760	797	832	860	885	912	955	995	1030	1063	1092
	650	778	825	864	900	932	963	987	1035	1078	1115	1152	1182
	700	838	888	932	970	1002	1035	1064	1114	1157	1200	1240	1275
	750	897	952	997	1038	1075	1105	1139	1193	1242	1284	1328	1364
	800	958	1012	1062	1108	1146	1180	1214	1264	1323	1372	1418	1455
n = 1.41	For p_s = 12.5, p_c =	33.1	45.5	58.8	73.1	88.3	103.0	120.6	156.5	194.0	234.0	277.0	321.0
	when T_s = 600, T_c =	735	783	826	864	897	927	956	1013	1055	1095	1135	1170
	650	795	847	893	936	972	1003	1033	1097	1140	1187	1228	1268
	700	857	912	962	1008	1048	1080	1113	1180	1228	1278	1320	1363
	750	917	976	1030	1077	1122	1157	1194	1265	1316	1368	1415	1462
	800	977	1142	1098	1149	1195	1232	1273	1348	1402	1458	1510	1558

It has already been shown in the previous chapter how the cyclic efficiency and consequently also the thermal efficiency of an engine depends upon the compression pressure. It has also been shown that there are commercial limits to the compression pressure due to pre-ignition of charge. As regards pre-ignition of charge, which is to be distinguished from back firing or explosions in the exhaust pipe, the greater danger of this exists with fuel mixtures high in hydrogen. Hydrogen, next to acetylene, possesses the lowest ignition point of any of the gaseous fuels commonly employed. Hence illuminating gas with about 45 per cent by volume of hydrogen is much more liable to pre-explosion under the temperature of compression than a producer gas with 15 per cent. Hence we may use a higher compression pressure in the case of the latter gas, and may expect, and actually obtain, a higher thermal efficiency in practice. With gases very rich in CO and low in hydrogen there is little danger of pre-ignition even up to the commercial limit of high pressures. Lucke estimates that for every 5 per cent of hydrogen that the gas contains 15 pounds should be subtracted from the otherwise allowable compression pressure. In general it should be borne in mind, and this applies to all fuel mixtures, the better and more effective the cooling of the cylinder, the higher can be the compression without danger of pre-ignition. Anything which draws down the temperature during compression, as water injection, is also favorable to the same thing. A case in point is Banki's method of water injection with gasoline as fuel, by which means compression pressures could be employed which gave thermal efficiency results equal to the best obtained on lean gases.

Premature explosions are sometimes directly due to faulty design of the combustion chamber. Any projecting point or edge in the chamber which cannot be effectively water-cooled may become red hot under compression, and thus locally raise the temperature high enough to cause pre-ignition. This action is so certain that it has been proposed to use it as a method of ignition by placing a projection on the piston face. It was found that while this scheme would work, it was not susceptible of control, and was abandoned.

The following table shows the safe compression pressures in use with the fuels commonly employed, as given by Lucke. The

second column gives the percentage of clearance required to produce the pressure p_c in terms of the piston displacement. This column has been computed assuming $p_s = 13$ lbs. absolute and $n = 1.35$.

Fuel	Compression Pressure Lbs. by gage	% Clearance in terms of Piston Displacement
Gasoline:		
Auto-engines with carbureters, cooling not	45 – 95	
very effective, high speeds	Ave., 65	35
Gasoline:		
Stationary, slow-speed engines, more effective cooling but usually less simple com-	60 – 85	
bustion chamber	Ave., 70	32
Kerosene:		
Hot bulb injection and ignition	30 – 75	35 – 40
Kerosene:		
Previously vaporized in devices not requir-	45 – 85	
ing a vacuum	Ave., 65	35
Natural Gas	75 – 130	
Natural Gas:		
Average for large and medium engines	Ave., 115	22
Illuminating gas	60 – 100	
	Ave., 80	26
Producer Gas	100 – 160	
Producer Gas:		
In large engines water-cooled on pistons		
and valves	Ave., 135	20
Blast Furnace Gas	120 – 190	
	Ave., 155	17

(c) THE COMBUSTION LINE. — The shape of the combustion line depends primarily upon the interrelation of three things: composition of charge, point of ignition, and piston speed.

For every fuel there is a certain fuel-air mixture which gives the greatest rate of flame propagation, *i.e.*, the most rapid combustion. Any further admixture of neutral gases, whether these be air or burned gases, results in a slower combustion, until there comes a time when ignition fails. Suppose, therefore, that in any given engine with full throttle, constant speed, and proper igni-

tion, we obtain diagram, Fig. 4–2, *a*. Now if the throttle is partly closed, making the dilution of the charge by the burned gases greater than before, card, Fig. 4–2, *b*, results. Further closure of the throttle makes the combustion still slower, Fig. 4–2, *c*. Similar effects would have been obtained with full throttle if the proportion of air in the fresh charge be seriously increased.

For proper combustion the time of ignition should be so chosen that the combustion line is vertical or nearly so. This means that every different fuel mixture and every different piston speed will have its own proper point of ignition. For that reason the ignition apparatus in every engine should be made adjustable, because the only way to determine the proper time is by trial. For small gasoline engines this adjusting may be done by ear, since the proper point of ignition corresponds nearly with the highest speed, for a given throttle position. For more accurate work the indicator, preferably with a constant-speed drum motion, should be used. It will in general be found when ignition is right that it occurs some time before the piston has reached the dead center, the amount of this lead depending upon the mixture and the piston speed as above stated. What improperly timed ignition results in is shown in a diagram, Fig. 4–3, given by Clerk. With proper ignition the normal diagram is indicated by *a*. As the time of ignition is made later and later, cards *b*, *c* and *d* result, in the last case flame propagation starting so

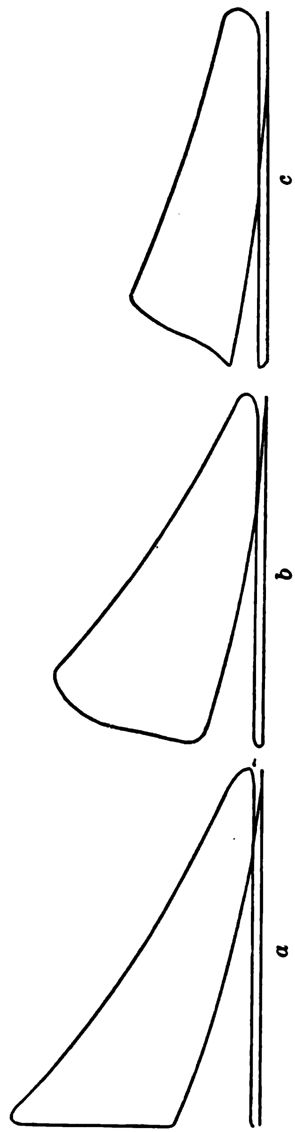


FIG. 4–2.

late that it barely overtakes the piston before the end of the stroke.

If instead of changing the time of ignition the piston speed had been increased, effects very similar to those of Fig. 4-3 would have been obtained.

Thus the dependence of the shape of the combustion line upon the three factors mentioned at the outset is clear.

The maximum pressure attained during combustion depends upon the heating value of the charge. With all conditions favorable, the maximum pressure should be reached at or before one-tenth stroke. The rich fuel mixtures, those for illuminating gas, natural gas, gasoline, etc., show rapid combustion. They

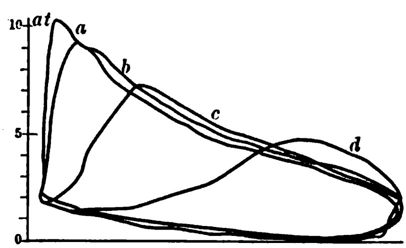


FIG. 4-3.

cannot in general stand high compression, and the ratio of $\frac{\text{maximum pressure}}{\text{compression pressure}}$, simply called the pressure ratio, is usually high, between 3 and 5. The naturally leaner fuels like producer gas and blast furnace gas ignite better when highly compressed, but on account of the generally low heating value of their fuel mixtures the pressure ratio for these gases is usually less, between 1.5 and 3.

The maximum explosion pressure p_x, Fig. 4-1, even assuming complete combustion at constant volume, is in no actual case as high as that computed on theoretical grounds for the heat received by the cycle. There may be several reasons for this. One is undoubtedly the loss of heat to the jacket water during explosion. This amount of heat is a dead loss since it does not even enter the cycle. It is something, however, which cannot be avoided, since cooling is a necessity for other reasons. Another is due to the undoubted fact that the specific heat of the gases increases

with the temperature. We can not, however, as yet mathematically gage this effect, because nothing definite is known of the law of increase.*

Another theory to account for the failure to realize theoretically maximum pressure assumes that combustion is not complete, that is, that not all of the heat of the charge is liberated when the piston starts forward. This results in after-burning, which will be considered later. It is quite likely that in most cases these three things combine to keep the observed pressure below that calculated.

Assume that the combustion takes place at constant volume, and let p_x and V_c, Fig. 4–1, be the pressure and volume at the end of the explosion.

Then $$p_x = \frac{p_c T_x}{T_c} \text{ and } T_x = \frac{p_x T_c}{p_c}$$

If the combustion line is other than vertical, as indicated by dotted line in Fig. 4–1, let p_x', T_x' and V_x' be the data for the end of the combustion. The above equations then become

$$p_x' = p_c \frac{T_x' V_c}{T_c V_x'} \text{ and } T_x' = T_c \frac{p_x' V_x'}{p_c V_c}$$

If in the above equations the values of p_x or $p_{x'}$ are taken from actual diagrams, the equations for T_x or T_x^1 will give real temperatures. But if p_x should be computed from one of the theoretical diagrams of the previous chapter, then the value of T_x should be multiplied by a factor which expresses how much the *real* value of p_x falls below the *theoretical* value of p_x due to imperfections of combustion. This factor is approximately equal to the ratio

$$\frac{\text{Indicated thermal efficiency}}{\text{cyclic efficiency}}$$

An idea of the maximum pressures p_x or p_x' existing in the cycle may be gained by considering that in most cases the ratio $\frac{p_x}{p_c}$ can be made equal to 3. Turning to the table, page 88, we

* See Chapter X.

see that for $n = 1.35$, and $r = 6$, p_c is equal to about 140 lb. absolute. Thus p_x would be about $3 \times 140 = 420$ lb. The temperature T_x, assuming $T_s = 700$ degrees, which for the same values of n and r, makes $T = 1114$, would then be about

$$T_x = \frac{420 \times 1114}{140} = 3342^\circ \text{ F. absolute.}$$

The following figures give a few of the characteristic diagrams for various fuels.

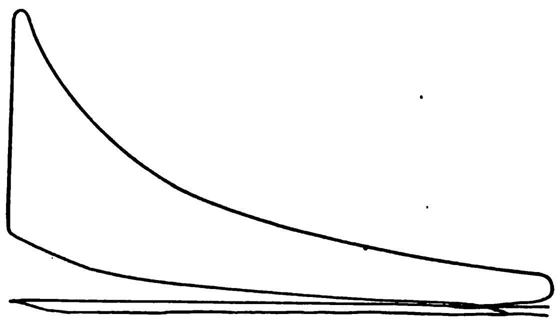


FIG. 4–4.

FIG. 4–4. From Struthers-Wells hit-and-miss engine. $11\frac{1}{2}'' \times 18''$, 30 H.P., 200 r.p.m., natural gas. Card good throughout, compression 80 lb., max. pressure, 320 lb. Pressure ratio $\frac{335}{95} = 3.5$.

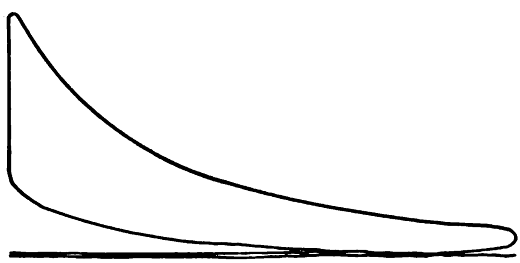


FIG. 4–5.

FIG. 4–5. From Struthers-Wells automatic engine, $16\frac{1}{2}'' \times 22''$, 150 H.P., 200 r.p.m., natural gas, full load. Card good, compression 96 lb., max., pressure 330 lb. Pressure ratio $\frac{345}{111} = 3.1$.

FIG. 4–6. From Stockport engine. Given by Clerk, the Gas and Oil Engine, p. 321. 9¾" x 17", 9 h.p., 182 r.p.m. Illuminating gas. Compression 90 lb., max. pressure 270 lb. Pressure ratio = 2.7.

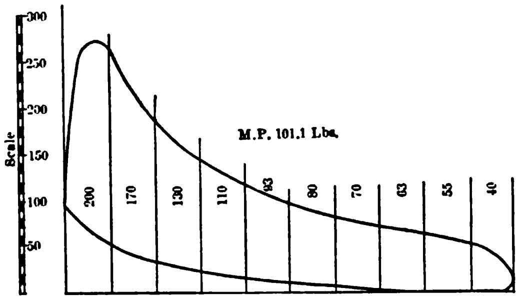


FIG. 4–6.

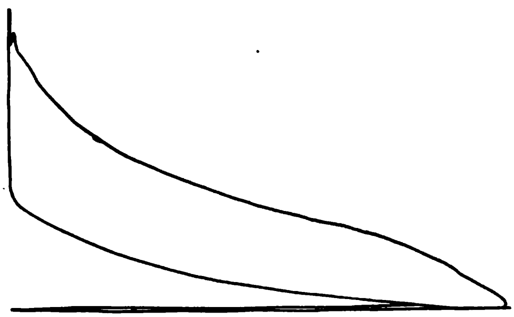


FIG. 4–7.

FIG. 4–7. From Hornsby-Akroyd kerosene engine. 6 H.P., 225 r.p.m., Card at about ¾ load. Hot bulb vaporization, hence pressures low. Compression 45 lbs., maximum pressure 116 lb. Pressure ratio = $\frac{131}{60}$ = 2.2.

FIG. 4–8. Card from a gasoline engine given by Lucke, Gas Engine Design, p. 71. Vapor prepared outside cylinder. Compression 80 lb., maximum pressure 372 lb. Pressure ratio $\frac{387}{95}$ = 4.07.

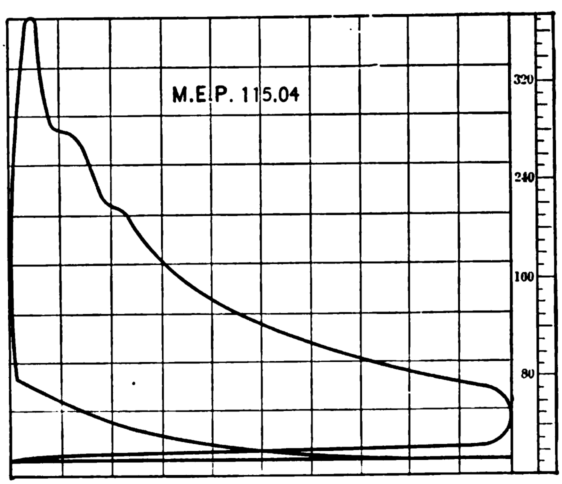


FIG. 4–8.

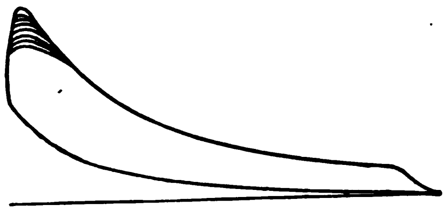


FIG. 4–9.

FIG. 4–9. Taken from Koerting engine. 700 H.P. Blast furnace gas. Max. pressure 242 lb., compression pressure 127 lb., Pressure ratio = $\frac{257}{143}$ = 1.8.

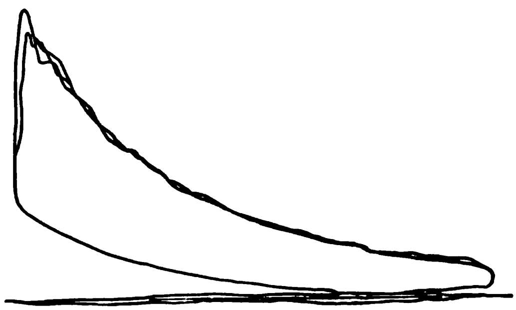


FIG. 4–10.

FIG. 4–10. From American Crossley producer gas engine, given by Langton. 18½ x 24″, 65 K.W., 200 r.p.m. Hit-and-miss governor. Gas rather high in H and low in CO, hence compression low. Compression 83 lb., maximum pressure 248 lb. Pressure ratio $\frac{263}{98}$ = 2.7.

The following two diagrams show abnormal conditions:

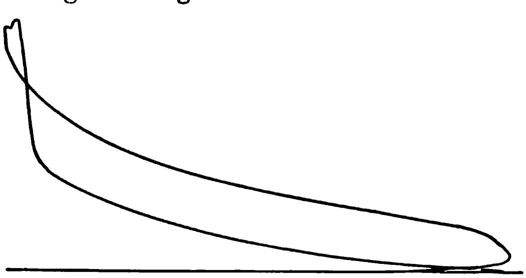


FIG. 4–11.

FIG. 4–11. Pronounced case of pre-ignition in 6 H.P. Hornsby–Akroyd kerosene engine due to too high compression. This was cured by the addition of a little water to the charge.

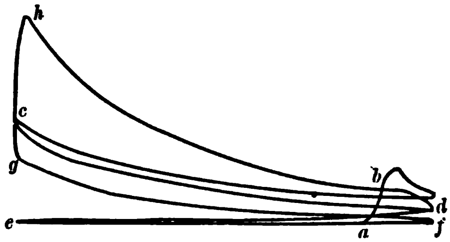


FIG. 4–12.

FIG. 4–12. Case of back-firing as distinguished from pre-ignition. Power, Oct. 15, 1900. Explosion in the suction pipe at *a* during the suction stroke. Observed in an engine which attempted scavenging by means of oscillations of burned gases in the exhaust pipe. Back-firing more often occurs in the exhaust pipe due to accumulation of unburned gas.

(*d*) THE EXPANSION LINE. — The expansion line, like the compression line, may be taken to follow a general law $p\,v^n =$ constant, in which n is rarely equal to $\gamma = \frac{C_p}{C_v}$, for the burned gases. Assuming that combustion is complete when the piston starts forward, the loss of heat to the jacket during expansion should cause the expansion line to lie below the adiabatic, that is, n should be greater than γ. (It should be remembered in this connection that leaky pistons and valves cause an increase in the true value of n.) Now it is very often found that the expansion line falls off much more slowly than this, coinciding now and then with the adiabatic, but very often lying between this and the isothermal, that is, n is less than γ. The explanation of this phenomenon has been held to be an evolution of heat along the expansion line equal to or exceeding the loss of heat to the jacket during this period. Several theories have been advanced to explain this.

The older machines of Lenoir and Hugon showed values of the exponent n for the expansion line between 1.4 and 1.6, while the earlier Otto machines showed values approximating 1.3. To explain this, Otto, and after him, Slaby, at least for a time, advanced the *theory of stratification.* It was supposed that the charge of a 4-cycle machine could be so arranged as to have practically nothing but burned gases against the piston, next practically air, then a layer of poor mixture, and finally near the igniter the fuel mixture in its full strength. It was further

assumed that this arrangement was disturbed but little during compression. Hence ignition was sure; but as the combustion progressed and reached the leaner layers of mixture, it became more and more slow, was not completed when the piston started forward, and was continued along the expansion line, showing a so-called after-burning. Thus Otto attempted to explain the somewhat slower rise of the combustion line and the absence of the serious shock at the moment of explosion in his gas engines. In the fight against the Otto patent, tests were made by the Deutz Company, by Dewar and by Teichman, all of which seem to support Otto's claim of stratification. Slaby defended it vigorously. It has, however, been pretty clearly shown to-day that, while stratification is not at all impossible, in fact it is not easy to get a uniform mixture, it cannot have any marked effect upon economy or performance; that is, the diagrams would not be very far different. The opinion of the best writers of to-day leans toward the requirements of most uniform mixture and rapid combustion. Clerk has attempted to show that some heat is "suppressed" during the combustion period in every gas engine and released along the expansion line, although to attempt to express these quantities in thermal units would seem superfluous in view of the fact that we know nothing definite of the variation of specific heat at high temperatures. It would seem, therefore, that after-burning, if it exists, is not a peculiarity common to Otto engines only. It is merely evidenced more strongly in the Otto diagrams by the fact of higher piston speeds and porportionately smaller enveloping surface, giving less time and opportunity for heat losses along the expansion line, as compared with gas engines before Otto's time. Hence, as Clerk puts it, the slow dropping of the expansion line is not the cause of the greater economy of the Otto engine, but rather the effect and evidence of it.

The second explanation of the supposed after-burning rests on the so-called *dissociation theory*. It has been shown by Bunsen that a composite gas breaks up into its elements when the temperature exceeds certain limits. Conversely, chemical combinations, such as combustion, can no longer take place when this temperature limit is reached. This theory applied to the gas engine would mean that at the inner piston position, if the temperature rises to the limit, combustion no longer takes place,

but just as soon as the piston starts forward, resulting in a drop of both temperature and pressure, combustion again ensues and is, by the same method, continued along the expansion line until no combustible remains. Clerk strongly leans to this view of the matter. But it has been pretty definitely shown that the temperatures in gas engines rarely exceed 28–3000 degrees Fahrenheit, and that these temperatures are below the dissociation limit. It has also been pointed out by Schöttler that if this theory holds good the expansion line should be an isothermal as long as there is combustion. It may therefore be concluded that dissociation plays but a small part in the combustion phenomena found in gas engines.

Witz, in some tests made with an engine whose piston speeds could be varied, found that in each case after-burning occurred, but that the combustion was the more rapid and the maximum pressure the higher (that is, after-burning the less noticeable), the greater the piston speed and the warmer the jacket walls. Thus he concludes that after-burning largely depends upon the influence of the walls. Slaby, corroborated by E. Meyer, on the basis of other tests, has tried to show that these conclusions of Witz are not generally applicable, but on weighing the evidence, and in the light of later achievements, it must be concluded that they and the principle based upon them, *i.e.*, rapid combustion of the leanest possible mixture at the greatest possible piston speed, at least represent a step in the right direction for gas engine economy.

From the above it is quite evident that none of the theories advanced explain satisfactorily all of the phases of the question of after-burning, the occurrence of which must be held as proven, especially in the light of later tests. Schöttler, indeed, has offered another explanation for the so-called abnormal position of the expansion line by showing that a natural solution of the question may be found in the variation of the specific heat of the expanding gases with temperature. Some figures quoted by him show that this may be the case, but before the idea of after-burning can be dispensed with, a great deal more experimental work is needed and desirable.

The requirements for best efficiency of combustion and expansion have already been briefly explained. In a little greater detail they are as follows:

1. Highest possible compression pressure before ignition. The effect of this is, (*a*) less admixture of burned gases to the fresh charge, (*b*) less loss to jacket because smaller volume is involved, (*c*) greater mean effective pressure, (*d*) greater ease of ignition of charge.

2. Pure and uniform mixture and rapid combustion to avoid after-burning. The bad effect of after-burning is due to the great jacket loss along the expansion line, and its effect has been compared by Koerting to that of a leaky valve in a steam engine.

3. Avoid external cooling. This of course cannot be entirely eliminated. But since the amount of heat lost by cooling is a function of both time and superficial surface, this requirement calls for high piston speeds and a form of cylinder in which the ratio $\frac{\text{superficial surface}}{\text{volume}}$ is the smallest possible.

To return to the expansion line, at the moment the exhaust valve opens, the gases have expanded to a volume V_y, with a pressure p_y, and a temperature T_y, see Fig. 4–1. We may write

$$p_y = p_x\left(\frac{v_c}{v_y}\right)^n, \text{ and } T_y = T^x\left(\frac{v_c}{v_y}\right)^{n-1}$$

The ratio $\frac{v_y}{v_c}$ is the real ratio of expansion. The expressions for the value of p_y and T_y for the constant-pressure cycle are analogous; care should be taken, however, to use the proper ratio of expansion, which in this case is quite different from the ratio of compression. In the case of the Otto cycle the ratio of expansion is in most instances nearly as great as the ratio of compression r, and we may therefore write, from the above expression for p_y

$$p_y = \frac{p_x}{r^n}$$

It appears from this equation that, with p_x remaining the same, the terminal pressure decreases as the compression increases. In practice, however, the terminal pressure in most cases shows an increase as compression is increased, due, no doubt, to the fact that p_x also increases with compression, unless the mixture is made correspondingly leaner.

(*e*) The Exhaust Stroke. — The velocity of efflux of gas at the instant the exhaust valve opens is very high, approximating 25–3500 ft. per second. The valve should start to open at about

$\frac{9}{10}$ out stroke, and the opening should be of such size that equalization of pressure is practically established by the time the outer dead center is reached. Too small an opening means increased lost work due to higher back pressure, higher mean cylinder temperatures, and less cylinder capacity. To get some idea of the charging and discharging operations of the four-cycle cylinder it is best to take the so-called loop card with a weak spring, say

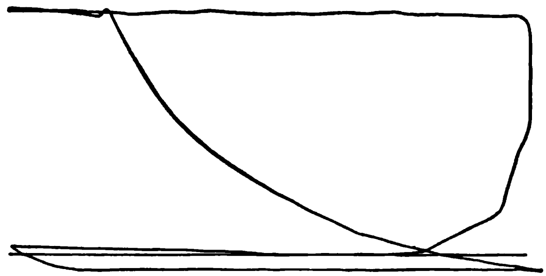


FIG. 4–13.

10–30 lb. scale, with stop attached. Besides giving a measure of the charging work, the so-called fluid friction, this also often reveals defects in the valve mechanism and sometimes curious variations in the exhaust line, which, however, are nearly always due, not to the engine, but to the exhaust piping. The ideal exhaust line should drop quickly nearly to atmosphere and remain so throughout the stroke. Fig. 4–13, from a $16\frac{1}{2} \times 22$

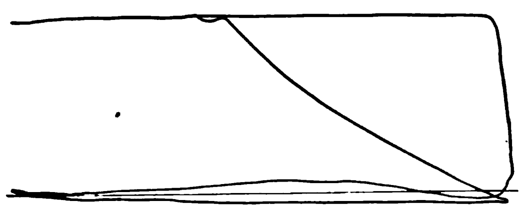


FIG. 4–14.

Struthers-Wells automatic engine on natural gas at full load, shows a nearly ideal exhaust line and in fact a very good loop card. Many loop diagrams, however, show a vacuum at the beginning of the exhaust stroke. This is undoubtedly due to the inertia of the initial gas column once set in motion. In some cases the indication of a vacuum is only very slight, in others it may last for half the stroke. In fact some builders have tried to

utilize the phenomenon to help scavenge the cylinder, but nothing has resulted from it because so many accidental conditions are apt to interfere with the regularity of its occurrence. Fig. 4–14 from a 6 horse-power Hornsby-Akroyd oil engine shows the occurrence of the vacuum very plainly. Owing to some peculiarities in the exhaust pipe, it sometimes happens that the masses of exhaust gas in the pipe are set into vibrations which cause a recurring vacuum in the exhaust line. Fig. 4–15, given by Güldner, shows a diagram from an Otto engine in which this occurred.

The absolute size of the loop, *i.e.*, the fluid friction, varies with the method of regulation, and, in a hit-and-miss engine, with the load on the engine. The effect of the method of regulation will be discussed in a later chapter. In a hit-and-miss engine at low loads the cylinder is cooler than at high loads, and the weight of gases displaced is greater, hence we may expect a greater area of loop than at normal loads. Humphrey, in a test of a 400

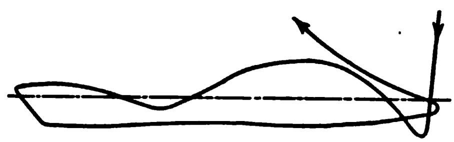


Fig. 4–15.

horse-power Crossley engine, determined a fluid friction loss of 15 horse-power at full load, which is $\frac{15}{400} = 3.8$ per cent. When the engine was taking in air only, this loss was 33 horse-power, but even assuming that there was no increased loss at say half load, the fluid friction would thus have been $\frac{15}{200} = 7.5$ per cent. It is found in general practice that fluid friction in hit-and-miss engines represents from 4–10 per cent of the engine power at full load, depending upon proper design.

3. **The two-stroke Otto Cycle.** Fundamentally there is no difference between the compression, combustion and expansion lines of the four-cycle and two-cycle types of engines, whether they operate on the constant-volume or constant-pressure combustion principle. The difference between the methods of operation is principally that the exhaust and charging actions of the four-cycle type are done in another way in the two-cycle, and that therefore the crank receives an impulse every revolution instead of every other. Fig. 4–16 shows an idealized two-cycle diagram.

The exhaust opens as before at *a*, the exhaust gases escape, charging commences, and is finished at *b*, where compression commences. That is the ordinary operation. There are some modifications, as for instance fuel is pumped in under high pressure along the compression line, or, as in the Diesel, air alone is compressed and the fuel injected only when the compression is completed; but these are special cases. It will be seen, therefore, from the diagram that in the ordinary case the exhaust and charging actions must be done during the time that the piston moves from *a* to the dead center position *c*, and back again to *b*. This time is short at best and extremely short under high speeds, and therein lies the whole difficulty of two-cycle operation.

The prime requirement of two-cycle operation is thorough scavenging of the cylinder of burned gases, for upon that depends

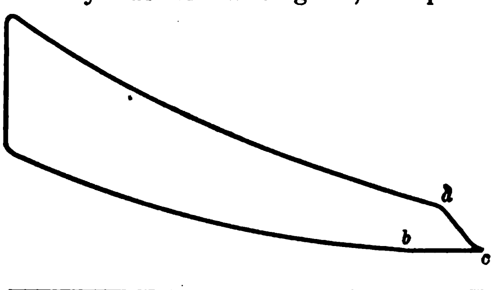


FIG. 4–16.

not only the volume of the fresh mixture that can be taken in, but also the explosibility of the charge. Too great a remainder of such gases not only seriously decreases the capacity of the machine, but it may even go so far as to prevent ignition altogether. Thus, as Güldner aptly says, the two-cycle stands or falls with the perfection or imperfection of the scavenging process. In the light of these facts it is comparatively easy to point out the requirement for good two-cycle operation.

I. The exhaust gases should be at approximately atmospheric pressure by the time the point *c*, Fig. 4–16, is reached. This reduces the volume of the gases remaining in the cylinder and reduces the work displacing them. For that reason the exhaust port should be of ample size, and this explains why the ring of ports, uncovered by the piston, is so much used. An exhaust valve as such is done away with.

II. Scavenging should commence somewhere between *a* and *c*. The scavenging agents used are

(*a*) Air; (*b*) fuel mixture.

The means by which the scavenging agent is furnished are

(*a′*) Separate pumps.

(*b′*) One end of cylinder or cross-head, used as pump.

(*c′*) Crank case used as pump.

Of these combinations, *a–a′* is undoubtedly the best. It is unquestionable that for thorough scavenging some excess of the agent should be employed, and for this only air and not fuel mixture can be used. Now only independent pumps admit of the obtaining of such an excess. It is plain, however, that it would not pay to construct independent pumps for all sizes of machines, hence they are restricted to large or at least medium powers. When independent pumps are not employed there is usually some deficiency of air. The designer, by proper design of combustion chamber and valves, is then compelled to do the best he can with the means at hand. Using the front end of the cylinder or designing the cross-head as a pump is better than using the crank case as such, mainly on account of less leakage, smaller clearance spaces and the possible better arrangement of valves. Fuel mixtures should under no circumstances be used for scavenging, except where cheapness of machine is the primary factor. Hence the small two-cycle machine which uses the fuel mixture as the scavenging agent, compressing it previously in the crank case. Such machines, however, present no true picture of two cycle operation or economy.

III. The scavenging agent should be of low pressure, and if possible of constant pressure. Low pressure is required to prevent the incoming air from piercing through and breaking up the mass of burned gases. The idea is to have the scavenging air shove these gases ahead of itself in a solid column. Constant pressure can be maintained only by the interposition of a reservoir between the pumps and the cylinder. Regarding the pressure of the scavenging air, however, a great deal depends upon valve and port construction, and speed of operation, hence nothing definite can be said to fit all cases.

4. **The Constant-pressure cycle.** At present this is only carried out as a four-stroke cycle with oil fuel in the Diesel engine. Fundamentally the modification in practice of the suction, compression,

expansion, and exhaust lines do not differ from those already outlined for the Otto cycle. The combustion line needs a little further attention.

The first cycle proposed by Diesel consisted of isothermal and then adiabatic compression, isothermal combustion and adiabatic expansion. This agrees with the Carnot cycle. Difficulties in the way of practical realization, however, lead to modifications of this proposed cycle until the actual cycle of to-day has little in common with it. In the first place, approximately adiabatic compression, such as is used in any gas engine, was substituted for the two compression lines originally proposed. The greatest change, however, is in the combustion line. In a lecture given by Diesel, a translation of which was printed in the *Progressive Age*, 1897, he lays down as the third requirement of his modified cycle "that the fuel must be introduced gradually into the air, which is compressed adiabatically to the combustion temperature, in such a manner that the heat generated by gradual combustion is absorbed in the so-called nascent state in consequence of a corresponding expansion, *i.e.*, by mechanically cooling off the gases, so that the period of combustion is going on constantly isothermally. It is evident that the fuel, in order to fulfil that condition, must be changed in its physical composition to a gaseous, liquid, or powdery form."

"That is to say, that through the combustion and during the same, no, or a relatively small, increase of temperature is caused, an idea which seems to be absurd after having heretofore always effected the increase in temperature by the combustion and during the same."

So far Diesel, even after the experimental stage of his engine had passed. It is quite evident that to approximate this isothermal combustion, it cannot, after ignition, be left to itself, but must be externally regulated to maintain the proper relation between temperature, pressure, and volume, as Diesel himself says. In the Diesel engine as at present constructed no such control is attempted, and it is hence difficult to see how isothermal combustion can be realized. Güldner, from indicator diagrams published by Schröter in 1897, accordingly found upon analysis that there was a decided temperature increase along the combustion line. The air was compressed to 600 degrees Centigrade (1132 degrees

Fahrenheit). At the full cut-off this had increased to about 1500 degrees Centigrade (2732 degrees Fahrenheit), and due to after-burning the maximum temperature was about 150 degrees Centigrade (270 degrees Fahrenheit) higher than this. The mean temperature of the four strokes was about 500 degrees Centigrade (932 degrees Fahrenheit). This in spite of the fact that the combustion line looked isothermal. In this case the maximum temperature was 2½ times that at the end of compression. The temperatures thus realized are *higher* than those found in engines using constant-volume combustion, in spite of claims to the contrary.

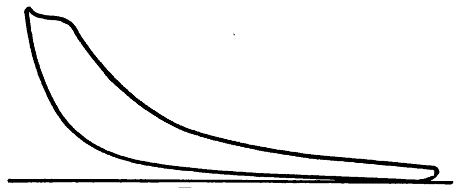


Fig. 4–17.

Fig. 4–17, from a German Diesel engine of late date, shows that the combustion is much nearer that at constant pressure than anything else. This is a step in the right direction, as it can be shown that isothermal combustion is the least favorable to best efficiency. (See previous chapter.)

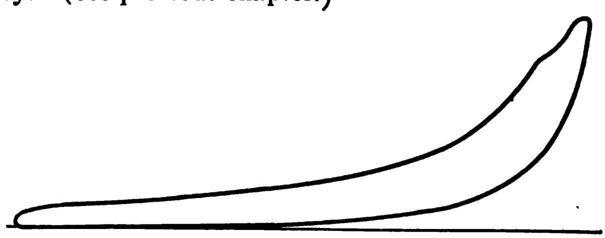


Fig. 4–18.

Fig. 4–18 shows a diagram published by the American Diesel Engine Company as late as 1904 or 1905. In the literature accompanying the diagram, the claim of isothermal combustion is still made. The combustion line certainly has that appearance, but data is unfortunately lacking to analyze the diagram. In view of the fact that the control of combustion in this engine has not changed very materially since 1897, it is safe to assume that the combustion is not isothermal, and it is perhaps to the advantage of the engine that it is not.

CHAPTER V

THE TEMPERATURE-ENTROPY DIAGRAM APPLIED TO THE GAS ENGINE

THE meaning of the term "Entropy" has already been explained in a previous chapter. It was also shown there what shape various pressure-volume diagrams assume when transformed to a temperature-entropy basis. It is proposed to show here mathematically, and graphically if possible, how this transformation is made.

Just as the *p-v* diagram, by areas developed, shows the amount of work done during various events of the cycle, the entropy diagram shows heat interchanges for the various parts of the cycle, and it is therefore a valuable aid in giving an insight into the thermal actions of the cylinder. Unfortunately the labor connected with the transposition of the *p-v* diagram of a gas engine to the entropy diagram is considerable, much greater than is the case for a steam-engine diagram. The reason is that in steam we have a medium whose properties are definite and unchangeable, and which are at once known when a single criterion of the state of the vapor is given. It is possible also to construct entropy tables for them which facilitate the work of transposition very much. On the other hand, a *p-v* diagram for a gas engine represents factors which hold for that diagram and no other. Such are the composition of the charge, the specific heats of the mixture, the exponents of the compression and expansion lines, etc., and all are important for the accurate determination of the entropy diagram. It may be said that for every different load on a gas engine these factors differ, and they will even change with accidental variation in engine operation at the same load. Hence the construction of entropy tables for gas mixtures, while possible, would be of use only when the conditions of operation

happened to fit the conditions which were assumed in the computations of such tables. That would not often be the case. Even a closely mathematical transformation of the *p-v* to the entropy diagram is subject to errors, which make many authorities skeptical as to the real value of the diagram. These errors may be grouped under two heads:

1. Errors due to transformation, that is, errors due to incorrectly measuring pressures and volumes from the *p-v* diagram, and mathematical errors of computation. All these may be kept below the required limit by sufficiently careful work. The original *p-v* diagram should be carefully enlarged and the measurements taken from this. The larger the scale, the better.

2. Errors due to assuming specific heat constant with changing pressure and temperature. How far this assumption is justified will be shown in a later chapter. We know that there is a variation, but the law is not definitely known for composite gases, as CO_2, and hence our assumption of constant specific heat results in a distorted entropy diagram. But this assumption is the best we can make with our present knowledge.

It should be clear from the above that a strictly graphical, and therefore time-saving, method of constructing the entropy diagram is out of question. The greatest stumbling block to the general applicability of such a method is apparently the variation of the specific heat with variation in composition of the charge. One of the best graphical methods assumes that the value of $\frac{C_v}{C_p - C_v}$ is always equal to 2.45 for the perfect gases. This law, however, appears to hold only for such gases as H, O, N and CO, whose $\frac{C_p}{C_v}$ is close to 1.41. For gases like superheated steam, or CO_2 the factor is considerably higher. Hence the value of $\frac{C_v}{C_p - C^v}$ at least should be fairly accurately computed even in using this graphical construction, the method for which will be explained later on.

Another point that should be mentioned is the fact that but few gas engine trials are sufficiently elaborate to furnish enough data for the mathematical determination of the entropy diagram. The following data should be known:

1. Composition and weight of fresh charge and burned gases.
2. Heating value of the fuel and of the charge.
3. Temperature at some point of the p-v diagram, preferably at end of suction stroke.
4. Stroke and clearance volume.
5. Index of the expansion and compression lines.

In the following will be given first the mathematical basis for the entropy diagram; this will be followed by the actual construction of such a diagram, and finally it will be shown how the same diagram can be obtained in a way mainly graphical.

The following mathematical exposition is due to Grover.* His demonstration of the general expression for entropy of a gas is especially clear.

Let H = quantity of heat in thermal units added to or subtracted from a mass of gas.

C_v = specific heat at constant volume.
C_p = specific heat at constant pressure.
T = absolute temperature.
p = absolute pressure in pounds per square foot.
v = volume in cubic feet.
J = mechanical equivalent of heat.
ϕ = entropy.

It may be remembered from the statements in Chap. II that

$$\delta\phi = \frac{\delta H}{T} \tag{1}$$

When heat is supplied to a mass of gas the volume, pressure, and temperature of the gas may vary simultaneously. Hence we may write the energy changes occurring under the conditions in the following general terms, following Grover:

Addition of heat may produce {Additional internal energy of the gas involving rise of temperature.} + {External effect of work done by the gas expanding between its containing walls.}

With the notation above given we may write this

$$\delta H = C_v\delta T + \frac{p}{J}\delta V \tag{2}$$

* Grover, Modern Gas and Oil Engines.

but $$\frac{pv}{T} = R = J(C_p - C_v) \tag{3}$$

hence $$p = J(C_p - C_v)\frac{T}{v} \tag{4}$$

Substituting (4) for p in equation (2)

$$\delta H = C_v\delta T + (C_p - C_v)\,T\frac{\delta v}{v} \tag{5}$$

and $$\delta\phi = \frac{\delta H}{T} = C_v\frac{\delta T}{T} + (C_p - C_v)\frac{\delta v}{v} \tag{6}$$

which is the general equation for entropy of a gas.

Now heat may be added to or subtracted from a body of gas under three conditions: (*a*) at constant volume; (*b*) at constant pressure; or (*c*) pressure and volume may change at the same time.

Case (*a*). Change of heat at constant volume.

Under this condition

$$(C_p - C_v)\frac{\delta v}{v} = 0 \tag{7}$$

and we shall have from (6) simply

$$\delta\phi = \frac{\delta H}{T} = C_v\frac{\delta T}{T} \tag{8}$$

Case (*b*). Change of heat at constant pressure.

From $pv = RT$

and $p\delta v = R\delta T$

we may derive

$$RT\frac{\delta v}{v} = R\delta T$$

or $$\frac{\delta v}{v} = \frac{\delta T}{T} \tag{9}$$

Substituting this value of $\frac{\delta v}{v}$ in equation (6) we have

$$\delta\phi = \frac{\delta H}{T} = C_p\frac{\delta T}{T} \tag{10}$$

Case (*c*). Simultaneous change of pressure and volume.

For this condition we may write

$$p\,v^n = p_1 v_1{}^n = \text{constant} \tag{11}$$

From $pv = RT$, we have

$$p = \frac{RT}{v} \tag{12}$$

Substituting (12) in (11)

$$RTv^{n-1} = \text{constant} \tag{13}$$

Differentiating (13)

$$\delta T v^{n-1} + T\,(n-1)\,v^{n-2}\,\delta v = O \tag{14}$$

$$\delta T v^{n-1} = -T\,(n-1)\,v^{n-2}\,\delta v \tag{15}$$

Divide by v^{n-1}

$$\delta T = -T\,(n-1)\frac{\delta v}{v} \tag{16}$$

or

$$\frac{\delta v}{v} = -\frac{\delta T}{T\,(n-1)} \tag{17}$$

Substituting (17) in general equation (6)

$$\delta\phi = \frac{\delta H}{T} = C_v\frac{\delta T}{T} - (C_p - C_v)\,\frac{\delta T}{T\,(n-1)} \tag{18}$$

To simplify this, write

$$\frac{C_p}{C_v} = \gamma, \text{ or } C_p = \gamma C_v \tag{19}$$

and substitute this value of C_p in (18)

$$\begin{aligned}\delta\phi = \frac{\delta H}{T} &= C_v\frac{\delta T}{T} - (\gamma C_v - C_v)\,\frac{\delta T}{T\,(n-1)}\\ &= \frac{\delta T}{T}\left(C_v - \frac{\gamma C_v - C_v}{n-1}\right)\\ &= C_v\frac{\delta T}{T}\left(1 - \frac{\gamma - 1}{n-1}\right)\\ &= C_v\frac{\delta T}{T}\left(\frac{n-\gamma}{n-1}\right)\end{aligned} \tag{20}$$

Restating these results we have

Volume Constant

$$\delta\phi = C_v\frac{\delta T}{T}$$

and entropy change is

$$\phi_2 - \phi_1 = C_v \log_e \frac{T_2}{T_1} \tag{21}$$

Pressure Constant

$$\delta\phi = C_p \frac{\delta T}{T}$$

and

$$\phi_2 - \phi_1 = C_p \log_e \frac{T_2}{T_1} \tag{22}$$

Change of volume and pressure, according to pv^n = constant.

$$\delta\phi = C_v \frac{\delta T}{T}\left(\frac{n-\gamma}{n-1}\right)$$

and

$$\phi_2 - \phi_1 = C_v \left(\frac{n-\gamma}{n-1}\right) \log_e \frac{T_2}{T_1} \tag{23}$$

As already stated, it is not easy to obtain tests which give sufficient data to construct the entropy diagram. One of the most complete was a test by Brooks and Steward, made twenty-three years ago at Stevens Institute. The engine tested developed about 7 horse-power on illuminating gas. Although several of the operating conditions of this test, especially the low compression pressure of 43.5 pounds, do not represent modern practice, the complete data furnished makes this test well adapted to the purpose in view.

I. Mathematical Construction of the Entropy Diagram.

All pressures and temperatures are absolute.

1. Composition and weight of fresh charge and burned gases.

(*a*) Composition of illuminating gas by volume.

H	CH_4	N	C_3H_6	CO	O	H_2O
.395	.373	.082	.066	.043	.014	.027

From this we compute the weight of a standard cubic foot of gas at .03882 pounds.

(*b*) Ratio of air to gas by volume = 6.63, from test.
Ratio of air to gas by weight = 13.78, from test.

(*c*) Volume and weight of the various gases per stroke.

The figures are given in the report in metric units. They are transposed as follows:

For one charge stroke the engine takes the following volumes:

$$\left.\begin{array}{l}\text{1.40 liters of gas}\\ \text{9.25 liters of air}\end{array}\right\}\text{at } 295^\circ\text{ C.} = \left\{\begin{array}{l}\text{.0494 cu. ft. of gas}\\ \text{.3265 cu. ft. of air}\end{array}\right\}\text{at } 533^\circ\text{ F.}$$

7.94 liters of burned gas at 683° C. = .2805 cu. ft. burned gas at 1261° F.

Now, —

.0494 cu. ft. of gas at 533° F. = .0457 cu. ft. at 493° F. which weigh .00178 lb.
.3265 cu. ft. of air at 533° F. = .3020 cu. ft. at 493° F. which weigh .02437 lb.
.2805 cu. ft. burned gas at 1261° F. = .1091 cu. ft. at 493° F. which weigh .00862 lb.

Hence total weight of charge = .03477 lb.

(*d*) Composition of exhaust gases.

From above composition of illuminating gas the theoretical ratio of air to gas by volume is 5.93. Hence excess coefficient is

$$\frac{6.63}{5.93} = 1.118$$

The weights of the products of combustion, therefore, from 1 pound of gas will be

$$\left.\begin{array}{l}1.940\text{ lb. } CO_2\\ 1.740\text{ lb. } H_2O\end{array}\right\}\text{from gas burned.}$$

.006 lb. N originally in gas.
9.490 lb. N from air used for combustion.

$$\left.\begin{array}{l}1.120\text{ lb. N}\\ .340\text{ lb. O}\end{array}\right\}\text{from excess air.}$$

From this the composition per cent by weight of the exhaust gases is

CO_2	H_2O	N	O
13.18	11.82	72.70	2.30

(*e*) C_p and C_v for the fresh charge and the exhaust gases are found by computation to be as follows:

For fresh charge, $C_p = .265$, $C_v = .191$, $\dfrac{C_p}{C_v} = \gamma = 1.39$

For burned gases, $C_p = .268$, $C_v = .196$, $\dfrac{C_p}{C_v} = \gamma = 1.37$

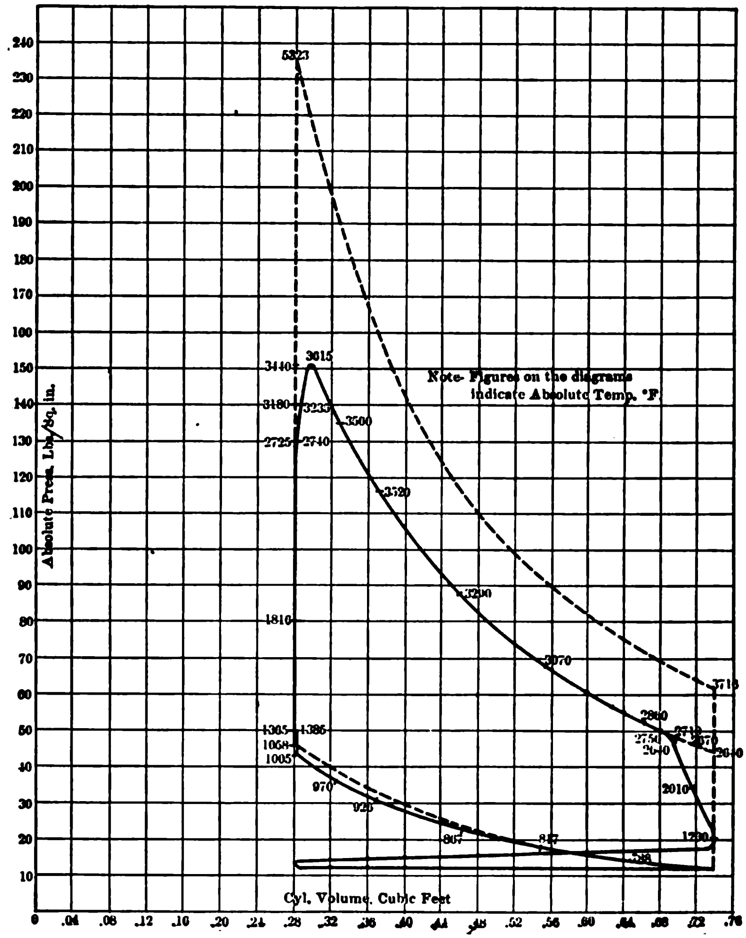


FIG. 5–1.

2. Heating value of the fuel is found by computation to be 617.5 B. T. U. per cubic foot.

Heating value of the charge as computed for the weight of gas in a charge is 29.06 B. T. U.

3. The absolute pressure at the end of the suction stroke, see diagram, Fig. 5–1, was 12.3 lb. Brooks and Stewart calculated the absolute temperature for the same point at 739 degrees Fáhrenheit. By using the equation

$$\frac{pv}{T}=\frac{p_1v_1}{T_1}$$

temperatures were computed for various points around the cycle as indicated on the card. In Fig. 5–1 the full line shows the actual card, enlarged from the original of Brooks and Stewart, while the broken line indicates the ideal cycle which receives the same amount of heat.

4. Stroke volume = .460 cu. ft.
   Clearance volume = .280 cu. ft.

   Total volume = .740 cu. ft.

5. Index for expansion and compression lines.

The index of the ideal card is 1.39 for the compression and 1.37 for the expansion line as computed under (1, *e*). For the real diagram it is often found that the index is not a constant for the entire line. For that reason each line should be divided into a number of parts and the index determined for each.

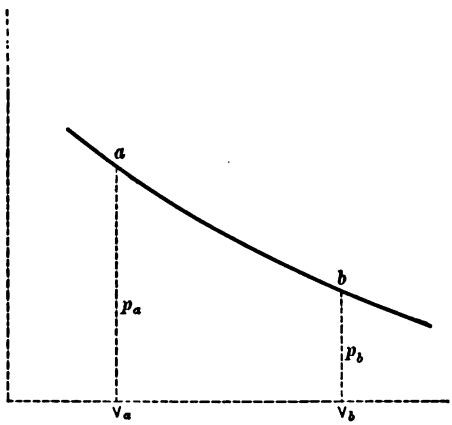


FIG. 5–2.

The method of doing this is as follows: To determine the index between any two points *a* and *b* on the expansion or compression line, Fig. 5–2, find the volumes and absolute pressures

for each point from the diagram. Then the exponent or index in the equation pv^n = constant is

$$n_{a-b} = \frac{\log p_a - \log p_b}{\log v_b - \log v_a}$$

In the case of the diagram under discussion this method when applied to the expansion and compression lines gave the following results:

Expansion line

Part of line	Index
54.6 to 68.2 lb.	n = 1.433
68.2 to 87.6 lb.	= 1.384
87.6 to 116.0 lb.	= 1.281
116.0 to 135.0 lb.	= 1.149

Compression line

Part of line	Index
12.3 to 18.1	n = 1.390
18.1 to 23.0	= 1.390
23.0 to 30.0	= 1.351
31.0 to 43.5	= 1.272

We are now ready to make the entropy computations. Since entropy difference from point to point and not absolute value of entropy is the important thing, we may call the entropy at any convenient point in the cycle equal to zero. We therefore assume the entropy at the end of the suction stroke equal to 0, and we will call the entropy difference between any two points $= \phi_1 - \phi_2 = \phi$.

Ideal Card. (See temperatures in Fig. 5–1.)

1. Compression line is adiabatic, hence entropy = 0 also at the end of this line.

2. Combustion line.

$$\phi = \phi_{231 \cdot 3} - \phi_{46 \cdot 6} = .196 \log_e \frac{5323}{1058} = .3166$$

3. Expansion line is adiabatic, hence entropy is not changed and = .3166 at end of this line.

4. Discharge line.

$$\phi = \phi_{12\cdot3} - \phi_{62\cdot8} = .196 \log_e \frac{739}{3716} = -.3166$$

This locates the end points of the combustion and discharge lines in the entropy diagram. But these lines are curves and hence several intermediate points on each line should be determined by similar computations. The diagram obtained by plotting temperatures and volumes of entropy above computed is shown in broken line in Fig. 5–3.

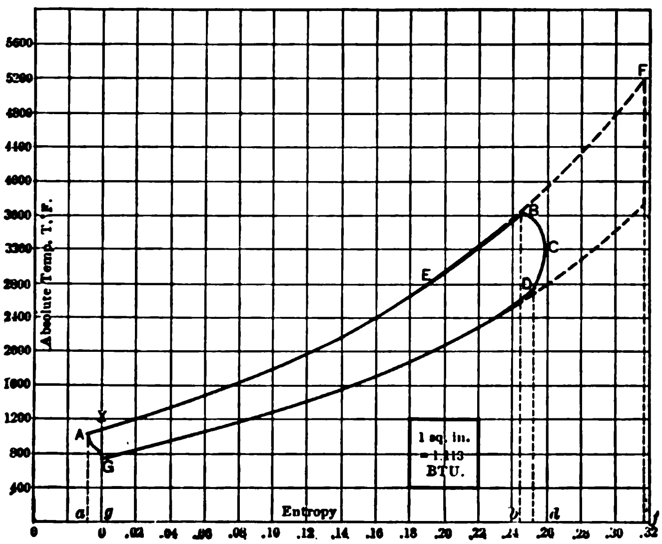


FIG. 5–3.

Actual Card. For the actual card we proceed in an exactly similar manner, using equations (21), (22), or (23) as the case demands. The computations are given below in detail. No further explanation seems necessary, except perhaps for the case where both pressure and volume change but the index of the curve is not known. Then none of the equations given directly apply.

Take the case of the diagram, Fig. 5–4. To determine the entropy at point *a*, find entropy at point *b*, using equation (21), and then add the entropy from *b* to *a* as computed from equation (22). Similarly for the point *c* on the exhaust line. Prolong

the expansion line and determine the entropy at d by equation (23) (index n must be known). Then subtract from this the entropy change due to the drop in temperature from d to c, according to equation (21).

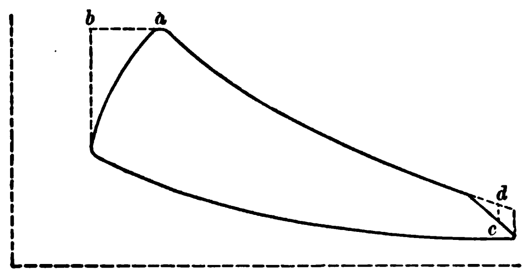


Fig. 5–4.

The following are the computations in detail:

Press. absolute	Temp.	Entropy at	Total Entropy
12.3	739	assumed = 0	0
18.1	817	adiabatic = 0	0
23.0	867	adiabatic = 0	0
30.0	926	$= .191 \frac{1.351 - 1.390}{.351} \log_e \frac{926}{867} = -.0014$	$-.0014$
43.5	1005	$= .191 \frac{1.272 - 1.390}{.272} \log_e \frac{1005}{926} = -.0066$	$-.0080$
60.0	1365	$= .196 \log_e \frac{1365}{1005} - .268 \log_e \frac{1385}{1365} = +.0591$	$+.0511$
80.0	1816	$= .196 \log_e \frac{1816}{1365} = +.0559$	$+.1070$
120.0	2740	$= .196 \log_e \frac{2725}{1816} + .268 \log_e \frac{2740}{2725} = +.0807$	$+.1877$
140.0	3235	$= .196 \log_e \frac{3180}{1816} + .268 \log_e \frac{3235}{3180} = +.1140$ (above 80 lbs.)	$+.2210$
151.3	3615	$= .196 \log_e \frac{3440}{1816} + .268 \log_e \frac{3615}{3440} = +.1376$ (above 80 lbs.)	$+.2446$
135.0	3560	$= .196 \log_e \frac{3060}{1816} + .268 \log_e \frac{3560}{3060} = +.1425$ (above 80 lbs.)	$+.2495$
116.0	3520	$= .196 \frac{1.149 - 1.370}{.149} \log_e \frac{3520}{3560} = +.0029$	$+.2524$
87.6	3290	$= .196 \frac{1.281 - 1.370}{.281} \log_e \frac{3290}{3520} = +.0033$	$+.2557$
68.2	3070	$= .196 \frac{1.384 - 1.370}{.384} \log_e \frac{3070}{3290} = -.0005$	$+.2552$

$$54.6 \quad 2860 = .196\frac{1.433-1.370}{.433}\log_e\frac{2860}{3070} = -.0019 \qquad +.2533$$

$$49.7 \quad 2750 = .196\frac{1.433-1.370}{.433}\log_e\frac{2750}{2860} = -.0007 \qquad +.2526$$

$$47.0 \quad 2640 = .196\frac{1.433-1.370}{.433}\log_e\frac{2710}{2750} - .196\log_e\frac{2710}{2640} = -.0053, \ +.2463$$

$$34.5 \quad 2010 = .196\frac{1.433-1.370}{.433}\log_e\frac{2670}{2750} - .196\log_e\frac{2670}{2010} = -.0566, \ +.1960$$

$$20.5 \quad 1230 = .196\frac{1.433-1.370}{.433}\log_e\frac{2640}{2750} - .196\log_e\frac{2640}{1230} = -.1550, \ +.0976$$

$$12.3 \quad 739 = .196\log_e\tfrac{739}{1230} = -.0978 \qquad -.0002$$

Having thus worked around the cycle, the error seems to be very slight in view of the fact that the exponent, n, for the prolonged expansion line has been assumed = 1.433.

Plotting these values of entropy and temperature finally results in the diagram shown in full line in Fig. 5–3.

Interpretation of the Entropy Diagram, Fig. 5–3.

In order to evaluate the diagram it is necessary to know the number of heat units per square inch. This is most easily obtained by multiplying one inch of temperature scale by one inch of entropy scale, and then multiplying the result by the charge weight per cycle, since the entropy diagram is drawn for 1 pound of charge weight. In this case we have:

Value of 1 square inch in B. T. U. = $800 \times .04 \times .03477 = 1.113$.

The individual areas of the original diagram were next gone over with a planimeter, and after multiplying each area by the square inch equivalent, the results were as follows:

	B. T. U.	% of Total Heat
1. Heat received during explosion, area $a\ A\ B\ b\ a$	18.086	62.24
2. Heat received during expansion, area $b\ B\ C\ D\ d$	1.002	3.44
3. Total heat received, as shown by diagram	19.088	65.68
4. Total heat supplied as calculated (see footnote)	29.316	
5. Difference in heat loss to Jacket and Radiation = area $d\,D\,C\,B\,E\,F\,f\,d$	10.228	34.89
6. Heat loss to exhaust, area $g\ G\ D\ d\ g$	13.422	45.78
7. Heat loss during compression, area $a\ A\ G\ g\ a$	.233	.79
8. Total heat lost per cycle	23.883	81.46
9. Indicated work, area $A\ B\ C\ D\ G\ A$	5.431	18.54
	29.314	100 00

The above results do not agree with those of Brooks and Stewart for the same test. Their results are given in the following table:

Heat in indicated work	17.0 per cent.
Heat loss in hot gases	15.5 per cent.
Heat loss in water jacket	52.0 per cent.
Heat loss in radiation	15.5 per cent.
Total	100.0 per cent.

The agreement as regards indicated work is fair. The rest of the figures of Brooks and Stewart are abnormal, in that the radiation loss is as large as the exhaust loss, and in that the jacket water loss is much too large. Brooks and Stewart themselves admit that the jacket water loss was not accurately determined. For that reason, too, the radiation loss cannot be found separately in the entropy analysis above given. If the jacket water loss had been accurately found, item 5 in the above analysis could have been separated into jacket water and radiation loss.

NOTE. — This quantity is greater than the latent heat energy in the gas = 29.06 B. T. U. per cycle, by the heat equivalent of the area $A\,X\,g\,a$ = .256 B. T. U.

II. Graphical Construction of the Entropy Diagram

The graphical method to be described is due to Prof. H. T. Eddy, and is by him explained in the Transactions of the American Society of Mechanical Engineers, Vol. 21, p. 275. It is based upon the following considerations:

Equation (6), p. 110, after integration may be written:

$$\text{Entropy difference } \phi = \phi_1 - \phi_2 = C_v \log_e \frac{T_1}{T_2} + (C_p - C_v) \log_e \frac{V_1}{V_2} \qquad (24)$$

Dividing equation (24) by $(C_p - C_v)$ we have

$$\frac{\phi}{C_p - C_v} = \phi' = \frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2} + \log_e \frac{V_1}{V_2} \qquad (25)$$

The problem then resolves itself into finding graphical representation for the quantities

$$\frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2} \text{ and } \log_e \frac{V_1}{V_2}.$$

The construction divides itself into two main parts:

1. The change of the pressure-volume diagram into a temperature-volume diagram, and

2. The change of the temperature-volume diagram so obtained into a temperature-entropy diagram.

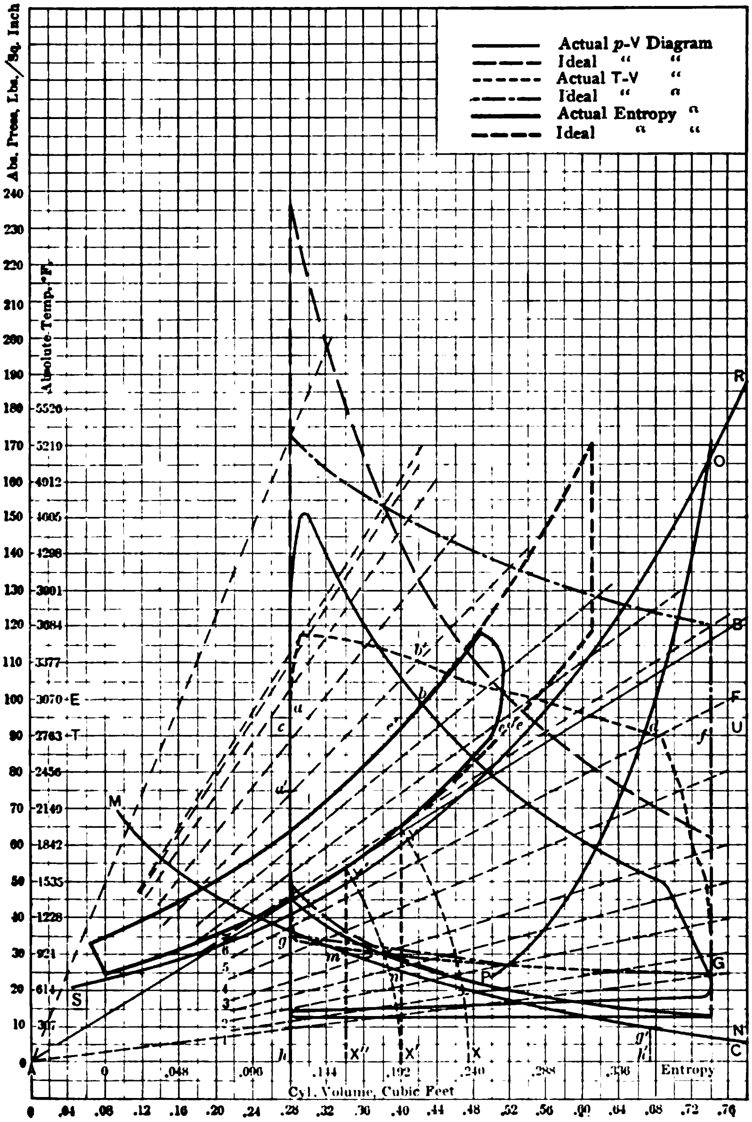


FIG. 5–5.

1. In Fig. 5–5 the actual and ideal p-v diagrams of Fig. 5–1 have been reproduced. Choose any convenient volume ordinate, in this case $V = .38$ cu. ft. has been taken, and from the points of intersection of the various pressure levels with this ordinate draw straight lines through the origin. These straight lines are constant-pressure lines in a temperature-volume field. Thus the line $A\ B$ in this case is the 60 pounds constant-pressure line. To see the reason for this consider the general equation

$$\frac{pv}{T}=R$$

This may be written

$$\frac{v}{T}=\frac{R}{p}, \text{ or } \frac{T}{v}=\frac{p}{R}$$

But $\frac{T}{v}$ is the tangent of any given angle, $B\ A\ C$, and hence $\frac{p}{R}$ is constant for any point along $A\ B$. The same holds for any straight line drawn through any other pressure level.

Next, to construct the temperature-volume diagram, consider any pressure level as $E\ F = 100$ lb. This cuts the real diagram in the points a and b. At a and b erect perpendiculars until they intersect the constant-pressure line = 100 lb. in the points a' and b'. These will be two points in the temperature-volume diagram. In the same manner the entire diagram may be outlined. The temperature scale, which up to this point has been arbitrary, may next be determined. We know that G, the lowest point on the temperature diagram, must represent the temperature in the cycle at the end of the suction stroke = 739 degrees Fahrenheit. This at once determines the temperature scale.

2. To obtain the graphical representation of the expression $\log_e \frac{V_1}{V_2}$ choose any convenient horizontal line as the zero. In this case the line $A\ C$ has been taken. With A as a center and any radius AX, draw the arc XY to cut any convenient line, as $A\ B$, passing through the origin. From Y draw the perpendicular YX'. Again with A as a center and AX' as radius, draw the arc $X'Y'$ to intersect with $A\ B$, and draw the perpendicular $Y'X''$. Where these perpendicular lines YX_1 and Y_1X'' cut any successive pair of equidistant horizontal rulings will be found two

points on the required curve. The horizontal rulings chosen in this case are marked serially on the ordinate $V = .22$ cu. ft. Thus m and n are two points on the curve MN sought. It is evident that the intersections of perpendiculars YX' and $Y'X''$ with any other two successive horizontal rulings might have been chosen as two points on the curve. This would have resulted merely in moving the curve bodily up or down on the field, the shape would have been exactly the same. For the same reason it is immaterial whether the line $A\ C$ or any other horizontal line is used as the base of construction. The resulting curve is in any case asymptotic to the vertical line $V = O$. Choosing any other set of equidistant horizontal rulings, nearer together or further apart, has the effect of making the curve rise slower or faster as the case may be. As will be seen later by inspection, this merely changes the position of the entropy diagram in the coordinate field but does not affect the final result.

To prove that the ordinates of the curve $M\ N$ represent the values of $\log \frac{V_1}{V_2}$, Professor Eddy proceeds as follows:

Let the volume $AX = V_o$, $AX' = V_1$, $AX'' = V_2$, $AX^n = V_n$. It can be shown by plane geometry that, with the construction used,

$$\frac{V_1}{V_0} = \frac{V_2}{V_1} = \frac{V_n}{V_{n-1}} = Z = \text{constant.}$$

or $ZV_0 = V_1, ZV_1 = V_2 \ldots\ldots\ldots\ldots\ldots\ldots ZV_{n-1} = V_n$

Hence

$$Z^2V_0 = V_2,\ Z^3V_0 = V_3 \ \ldots\ldots\ldots\ldots\ldots\ldots\ldots Z^nV_0 = V_n$$

Now let $Z = e^y$ in which e = Naperian base = 2.7 +, and y = distance between the equidistant horizontal rulings above assumed.

Then

$$e^y = \frac{V_1}{V_0},\ e^{2y} = \frac{V_2}{V_0}\ \ldots\ldots\ e^{ny} = \frac{V_n}{V_0}$$

and taking the logarithm of both sides of the last term, we have $n\,y$ = the ordinate at any given volume $V_n = \log_e \frac{V_n}{V_0}$. It is evident that V_o, the unit of comparison, may be arbitrarily chosen.

The next step is to obtain the graphical representation of the expression $\frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2}$. Considering the part $\log_e \frac{T_1}{T_2}$ by itself,

it is evident that the curve representing this may be constructed in the same manner as curve $M N$. But the coordinates are in this case the line of zero temperatures, $A C$, and any arbitrarily chosen vertical line, in this case, $V = .74$. The curve must be asymptotic to $A C$, but the choice of the other line of reference is unrestricted as it merely moves the curve bodily to the left or right. In this case equidistant vertical rulings having the same common distance as those used for curve $M N$ have been employed, hence curve $O P$ is a duplicate of $M N$.

The ordinates of this curve $O P$ must next be multiplied by the factor $\frac{C_v}{C_p - C_v}$. Professor Eddy, in order to make the construction graphical throughout, assumes that the value of this factor is 2.45 in all cases. The limits of accuracy regarding this assumption have already been pointed out. It is probably *not* sufficiently accurate for most cases, and hence a separate computation of this factor is necessary for every given case. This destroys a great deal of the value of the entire method, but enough is left to make the method much less laborious than the mathematical construction.

In the case under discussion, for the burned gases,

$$C_v = .196, \text{ and } C_p = .268; \text{ hence } \frac{C_v}{C_p - C_v} = \frac{.196}{.072} = 2.72$$

The ordinates of the curve $O P$ are therefore multiplied by 2.72, giving the curve marked $R S$.

By the aid of the curves $M N$ and $R S$, the entropy diagram may now be constructed in the following manner:

Take any temperature level as $T U$. This cuts the temperature-volume diagram of the real cycle in the points c and d, and the curve $R S$ in e. With a pair of dividers determine the ordinate $g h$ of the curve $M N$, $\left(\log_e \frac{V_1}{V_2}\right)$, corresponding to the volume of point c, and since this ordinate is positive, add it to the ordinate $f e$ $\left(\frac{C_v}{C_p - C_v} \log_e \frac{T_1}{T_2}\right)$ of the curve $R S$. This gives the point e' as one point of the entropy-temperature diagram. In the same manner, for the second point of intersection, d, of the temperature level $T U$, determine the ordinate $g' h'$, corresponding to its volume, and add it to the ordinate fe of the curve $R S$. This

gives e'' as a second point on the entropy diagram. By taking a sufficient number of temperature levels, the entire diagram may be closely outlined, as shown by the full line.

The same thing has been done for the ideal temperature-volume diagram, giving the ideal entropy diagram indicated in broken line.

The last step in the construction is the determination of the entropy scale, if this is desired. Determine by planimeter the area under the combustion line of the ideal diagram down to the line $T = 0$. In the original diagram this was found to be 28.32 square inches. Since the heat applied the cycle was 29.06 B. T. U., the thermal value of each square inch of area $= \frac{29.06}{28.32} =$ 1.026 B. T. U. Each inch of ordinate was equal to 614 degrees, hence the entropy scale per inch is $\frac{1.026}{614} = .00167$ for the charge weight of .03477 pounds. This is equivalent to an entropy scale of .048 per inch for one pound of charge weight.

The following table shows how closely the entropy diagrams obtained by the two methods outlined agree:

	Math. Method	Graph. Method
Max. Entropy of Ideal Cycle	.3166	.3158
Max. Entropy of Real Cycle.....	.2557	.2572

The agreement may be pronounced quite satisfactory. It is quite likely, however, that the lower part of the graphical entropy diagram will show discrepancies. These are in great part due to the fact that toward the lower end of the curve $R S$, the intersections with the horizontal temperature levels become less definite, impairing the accuracy. Another source of error may lie in the fact that the compression line has been constructed from the curve $R S$, which was itself constructed from burned gas data, while evidently the data of the fresh charge should have been used.

CHAPTER VI

COMBUSTION

1. The Perfect Gases. — The perfect gases are those which follow the general law:

$$\frac{pv}{T}=R \qquad (1)$$

where

p = pressure expressed in pounds per square foot.

v = volume in cubic feet.

T = absolute temperature.

R is the amount of work done by 1 pound of gas when heated 1 degree Fahrenheit, the pressure remaining constant at p pounds per square foot. R is thus a constant for any one gas, but differs for different gases.

The data for the gases of most use in gas-engine practice are those of atomic and molecular weight, density and weight per cubic foot. The following table gives these figures for some of the more important gases:

Gas	Atomic Weight	Molecular Formula	Molecular Weight	Density Air = 1	Weight per cu. ft. at 29.92″ Hg and 32°F
Hydrogen	1	H_2	2	.0692	.00559
Oxygen	16	O_2	32	1.106	.08921
Nitrogen	14	N_2	28	.971	.07831
Carbon Monoxide .	14	CO	28	.967	.07807
Carbon Dioxide ...	14.6	CO_2	44	1.529	.12267
Dry air	—	—	29*	1.00	.08072
Water vapor	6	H_2O	18	.623	.05020
Acetylene.........	6.5	C_2H_2	26	.915	.07251
Methane	3.2	CH_4	16	.554	.04464
Ethylene	4.7	C_2H_4	28	.974	.07809
Benzol	5.75	C_6H_6	78	2.695	.21758
Alcohol...........	6.50	C_2H_6O	46	1.601	.12958

*Only apparent value.

The weight of a cubic foot of any of the above gases under standard conditions may be found with sufficient accuracy by

dividing the molecular weight of the gas by the constant 359.* The weight of a cubic foot of CO_2 under standard conditions, for instance, is $\frac{44}{359} = 0.1225$ pounds, as computed by slide rule. This is sufficiently accurate for all practical purposes.

To find the weight of a cubic foot of gas under other than standard conditions, the formula

$$\frac{p_0 v_0}{T_0} = \frac{p_1 v_1}{T_1} \tag{2}$$

may be employed for a change of either pressure and temperature separately or a simultaneous change of both. The weight of a cubic foot is inversely proportional to the volume.

2. Combining Weights and Volumes, Combustion, Heating Value, Air Required.

COMBINING WEIGHTS AND VOLUMES. — All elements, when they do combine, unite only in certain fixed proportions, although there may be several proportions for any given pair of elements. Thus carbon forms two combinations with oxygen, CO and CO_2.

The gases also combine in definite volume proportions. The resulting volume is either the sum of the original volumes or is in a definite ratio less than this sum. It is necessary to remember merely that anything that can be said of molecules according to Avogadro's Law, applies with equal force to combining volumes.

Thus take the combination of C and O to CO. Always remembering to use the combining weights of the various elements we can write

$$C_2 + O_2 = 2CO$$

that is

$$1 \text{ vol. C} + 1 \text{ vol. O} = 2 \text{ vols. CO.}$$

Similarly,

$$2\,H_2 + O_2 = 2\,H_2O.$$

that is,

$$2 \text{ vols. H} + 1 \text{ vol. O} = 2 \text{ vols. } H_2O$$

and again

$$C_2 + 4\,H_2 = 2\,CH_4$$

or

$$1 \text{ vol. C} + 4 \text{ vols. H} = 2 \text{ vols. } CH_4.$$

The elements mostly concerned in combustion phenomena are carbon and hydrogen, together with the compounds carbon monoxide and the various hydro-carbons. The following table

* Derived, in connection with Avogadro's Law, from the fact that the kilogramme-volume of perfect gases is equal to 22.33 cubic meters under standard conditions.

shows the combustion formula, also oxygen and air required, for the first three of these combustibles:

Combustible	Combustion Formula	Combining Weights	Combining Volumes	Air required			
				Pounds per		Cu. ft. per*	
				Pound	Cu. ft.*	Pound	Cu. ft.*
H	$H_2 + O = H_2O$	1 lb. H + 8 lb. O = 9 lb. H_2O	2 vol. H + 1 vol. O = 2 vol. H_2O	34.78	.194	430.9	2.408
C to CO	C + O = CO	1 lb. C + 1.33 lb. O = 2.33 lb. CO	1 vol. C + 1 vol. O = 2 vol. CO	5.78	—	71.6	—
C to CO	$C + O_2 = C_2O$	1 lb. C + 2.66 lb. O = 3.66 lb. CO_2	1 vol. C + 2 vol. O = 2 vol. CO_2	11.57	—	143.4	—
CO to Co_2	$CO + O = C_2O$	1 lb. CO + .57 lb. O = 1.57 lb. CO_2	2 vol. CO + 1 vol. O = 2 vol. CO_2	2.48	.194	30.6	2.388

* Standard conditions, *i.e.*, 29.92 inches barometer and 32° F.

If the combustible be a hydrocarbon, the combustion of the carbon and the hydrogen in its composition can be treated separately and the results combined. Thus 1 lb. of CH_4 may be considered to consist of $\frac{3}{4}$ lb. of C and $\frac{1}{4}$ lb. of H.

Heating Value. — Every chemical change is accompanied by a thermal change, either positive or negative, *i.e.*, either heat is given out or it is absorbed during the change. In the case of the combustibles, the union with oxygen is accompanied by a very decided development of heat. The heat given off when one pound of any combustible is completely burned is known as its *heating value*, or *calorific power*. It is to be noted that the heating value of any combustible is constant, whether the combustible

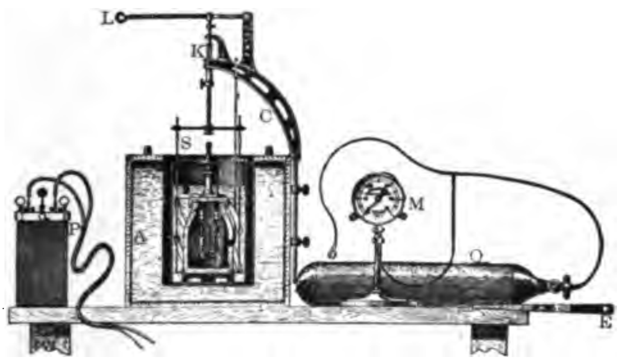


Fig. 6–1. — Mahler Bomb Calorimeter.

is burned in oxygen, theoretical amount of air, or an excess of air. The resulting temperature, the *calorific intensity* so called, is, however, different for each of these cases, as will be explained later.

The heating values of the simple combustibles like hydrogen and carbon can only be determined by means of the calorimeter; that of a complex fuel, like hydrocarbons, the various coals, etc., can be found either by the calorimeter or it may be computed with fair accuracy from its chemical composition.

For solid and liquid fuels the calorimeters used are mostly of the type of the Mahler bomb, or the Carpenter calorimeter; for gaseous fuels Junker's gas calorimeter holds the first place. The principle of any of these calorimeters is to transmit to water

the heat evolved by burning the fuel in oxygen or air, and from the temperature rise of the water to compute the heating value.

The Mahler Bomb Calorimeter, Fig. 6–1, is a strong steel or bronze vessel into which the finely powdered fuel is introduced, held in a small cup or crucible. Through the cover of this vessel or bomb two wires pass which at their lower ends are cross-connected by a fine iron wire which in turn dips into the powdered fuel. The bomb is charged with oxygen to a pressure of about 150 pounds. The oxygen can now be obtained commercially prepared in steel tubes, under pressure. The charged bomb is then placed in a vessel containing a known quantity of water; the two wires above mentioned are connected to a source

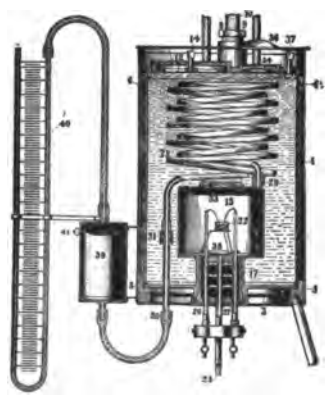


FIG. 6–2. — Carpenter Coal Calorimeter.

of electrical energy, *P*, which fuses the iron wire connecting their ends and fires the fuel. The water is kept thoroughly stirred by means of the apparatus *L K S*, and the temperature rise is carefully determined. The whole apparatus is carefully protected against radiation by an outer vessel, *A*. From the observed temperature rise, the weight of water, and such corrections as are necessary for radiation, heat of fusion of wire, etc., the heating value of the sample is easily computed. The drawbacks of the instrument are the labor of charging, and the fact that if the wire fails to ignite the coal all labor of weighing and charging is lost. The joint at the top, which must be tight against considerable pressure, also often gives trouble. The Mahler is applicable to both solid and liquid fuels.

The Carpenter calorimeter, Fig. 6–2, is of a different type. The fuel is powdered and held in the cup, 22, just as in the Mahler. The coal, however, is fired by the heat of a platinum wire, 15, which does not fuse, and is extinguished on the instant the coal fires. No correction is therefore necessary for the heat of the wire. The combustion chamber, 33, is kept charged with a steady stream of oxygen under a pressure not exceeding one-half pound. The hot gases of combustion pass up and down the spiral tube, 28, and finally escape at 41, practically at the tem-

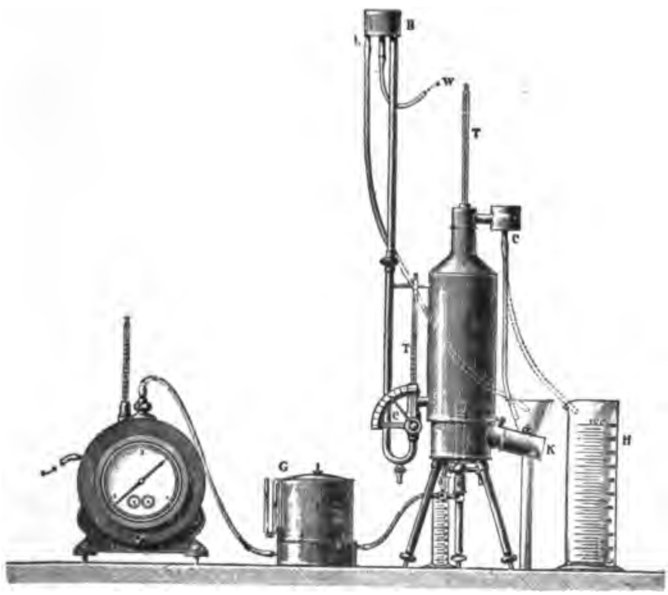


FIG. 6–3. — Junker's Gas Calorimeter.

perature of the surrounding air, the nozzle at 41 regulating the outflow velocity to produce this result. The calorimeter is completely filled to some height on the glass tube, 10, with water or kerosene oil so that the gases in their passage heat the fluid, expanding it up the tube 10. The amount of expansion gives a relative measure of the heating value of the fuel. It is only necessary to calibrate the instrument, *i.e.*, to obtain the amount of heat required to expand the fluid up the tube say 1 inch. This is quickly and accurately done by using carbon obtained by

burning and coking sugar. The calorimeter is subject to one correction only, that for radiation. It is encased in a bright nickel case to reduce radiation as far as possible, but this can not be entirely done away with. The correction is made by determining the distance the liquid level in the tube will fall, after the fuel sample is burned out, for the same length of time that the fuel burned. Care must be taken to have the temperature of the calorimeter slightly above the temperature of the room to start with, and if the temperature of the room has not changed during the time of burning and that of the determination for radiation, the fall in the water level is a true measure of the heat loss due to radiation.

For gas fuels, Junker's calorimeter, Fig. 6–3, is usually preferred. The gas to be tested is led first through the gas meter where its volume and temperature are determined, and second through the regulator g, which maintains any desired gas pressure during the determination. It is burned in air, with a burner of the Bunsen type. The heat evolved is transmitted to water, entering through the vessel b, and discharged through the vessel c, the gases escaping at or near the room temperature at K. The vessel b maintains a constant head of water on the calorimeter. The amount of water flowing is regulated at e. The water is measured in the graduate h. The water of condensation formed during the combustion is caught and measured at d. Temperatures of cold and hot water are measured at t and t' respectively. The apparatus is very simple, easily set up in any desired place, and the results obtained by its use are consistent and reliable. Its operation may be made continuous by furnishing suitable means for measuring the water flowing.

The heating value of carbon when burned to CO_2 is now definitely determined as 14647 B. T. U. per pound (8080 calories per kilo), that of carbon to CO at 4429 B. T. U. In actual practice the figures 14500 B. T. U. for C to CO_2, and 4500 B. T. U. for C to CO are easier to remember and accurate enough for most purposes. The combustion of C to CO is usually known as the incomplete combustion of carbon. The heating value of the combustion of CO to CO_2 has been found to be 4380 B. T. U.

Hydrogen gives for the *higher* heating value 62100 B. T. U., and for the lower value 52230 B. T. U. Hydrogen may burn

either to water vapor, or, if condensation occurs, to water. In the former case the heat necessary to maintain the water as vapor is lost, and hence we obtain the lower heating value 52230 B. T. U., the temperature being 212 degrees Fahrenheit before and after combustion. If condensation occurs, the heat in the water vapor is recovered and the higher value 62100 B. T. U. results.

In gas-engine practice the water resulting from the combustion of hydrogen almost always escapes as water vapor, and hence it is usual to employ the *lower* heating value in computations.

For the same reason we distinguish a lower and higher heating value in all complex fuels containing hydrogen to any considerable extent, and always use the *lower* value in computations.

If no direct determination of the heating value of a complex fuel can be made by means of a calorimeter, the heating value can in most cases be found with fair accuracy by computation, provided the chemical analysis of the fuel is known.

In the case of hydrocarbons the statement of its chemical formula gives at the same time the weight proportions of its chemical composition. The computation of the heating values of the hydrocarbon groups C_nH_n and C_nH_{2n} may be based upon the following considerations:

C_nH_n contains n atoms of C and n atoms of H. Hence the gram-molecule weighs $12n + n = 13n$ grams. Now 1 gram of C develops 32.29 B. T. U., and 1 gram of H 115.15 B. T. U. lower heating value. The heating value of C_nH_n is therefore

$$\frac{(12 \times 32.29)\, n + (1 \times 115.15)\, n}{(12 + 1)\, n} = 38.67 \text{ B. T. U. per gram}$$

This reduces to:

Lower heating value of C_nH_n = 17540 B. T. U. per lb.

C_nH_{2n} contains n atoms of C and $2n$ atoms of H, one gram-molecule therefore weighs $14n$ grams. The lower heating value of this family of hydrocarbons is therefore generally

$$\frac{(12 \times 32.29)\, n + (1 \times 115.15)\, 2n}{(12 + 2)\, n} = 44.13 \text{ B. T. U. per gram}$$

$$= 20016 \text{ B. T. U. per lb.}$$

It should be remembered, however, that these formulæ give approximate results only.

Slaby, quoted by Güldner,* in his Calorimetric Investigations, gives a formula based upon experience for heavy hydrocarbons, according to which:

Lower heating value = 1000 + 10500 y calories per cubic meter, where y = weight of a cubic meter of the gas in Kgs.

In English units this formula would be:

$$\text{Lower heating value} = [112 + 18880\,y] \text{ B. T. U. per cubic foot,} \qquad (3)$$

the cubic foot being under standard conditions, 29.92 inches Hg barometer and 32 degrees Fahrenheit, and y = weight of a cubic foot.

The following table of heating value of hydrocarbons is taken from Güldner, the figures being transposed to English units. The last two columns show for purposes of comparison the values as computed by Slaby's formula and the approximate formulæ first given:

Heating Values of Hydrocarbons

	Weight of cu. ft. in lbs. Standard	Density air = 1	Higher Heating Value per lb. B. T. U.	Lower Heating Value		Lower Heating Value per cu. ft Slaby's formula B. T. U.	Lower Heating Value Approximate Formula per cu. ft. B. T. U.
				per lb. B. T. U.	per cu ft. B. T. U.		
CH_4 ..	.04464	.554	23842	21385	952	952	
C_2H_2 ..	.07251	.915	21429	20673	1499	1479	1272
C_2H_4 ..	.07809	.974	21429	20025	1564	1584	1562
C_2H_6 ..	.08329	1.0367	22399	20434	1700	1682	
C_3H_4 ..	.11157	1.3819	20992	20009	2232	2238	
C_3H_6 ..	11699	1.4512	21224	19820	2317	2318	2341
C_3H_8 ..	12256	1.5204	21825	20039	2455	2424	
C_4H_8 ..	15599	1.9349	20912	19508	3032	3055	3120

In the case of the liquid hydrocarbons, the crude oils and their distillates, it is usual to compute their heating value directly from the chemical composition. Thus the lower heating value

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, 2d ed., p. 581.

of a Pennsylvania crude oil containing C 84.9 per cent, H 13.7 per cent, O 1.04 per cent, would be

$$= 14500\ C + 52230\left(H - \frac{O}{8}\right) \qquad (4)$$

$$= [14500 \times .849] + \left[52230\left(.137 - \frac{.0104}{8}\right)\right]$$

$$= 12310 \qquad + \qquad 7102$$

$$= 19412 \text{ B. T. U.}$$

The determination of the heating value of this oil by means of the calorimeter gave 19210 B. T. U., a difference of 1.05 per cent between this and the computed value.

In many instances, however, the agreement is not as close, and it should therefore be made a general rule to obtain actual calorimeter determinations whenever possible, and to resort to computation only when unavoidable.

Solid fuels, as coal, coke, wood, etc., may be treated in a similar manner. For hard coals the computation gives results which agree fairly closely with calorimeter determinations. As the amount of volatile matter in the coal increases, *i.e.*, as the hydrocarbons increase, there is less certainty of fair agreement, although the computed result is usually within 5 per cent of the true value. The general formula for a solid fuel may be stated as follows:

Heating value =

$$14500\ C + 52230\left[H - \frac{O}{8}\right] + 4000\ S - 1000\ H_2O \qquad (5)$$

where

C = percentage of fixed and volatile carbon.
H = percentage of hydrogen.
O = percentage of oxygen.
S = percentage of sulphur.

H_2O = percentage of water in coal as received, C, H and S being determined on the dry fuel, and then recomputed to the basis of fuel as received.

The following table shows the analyses of two coals and the heating values as found by calorimeter and by above formula. Both analyses are taken from Poole's "The Calorific Power of Fuels:"

Name	Kind	C	H	O	N	S	H_2O	Ash	Heating Value as received	
									By Cal.	Computed
Treverton..	Anthracite	90.66	1.73	.78	.001	—	.84	6.83	14029	13988
Carnegie...	Bituminous	77.20	5.10	7.22	1.68	1.42	1.45	5.93	13842	13367

It will be noted that the agreement in the results for the hard coal is good, in the case of the soft coal the computed result is 3.5 per cent smaller than that determined by calorimeter.

Attempts have been made from time to time to adapt formulæ to the approximate analyses of coal, *i.e.*, those in which only fixed carbon, volatile matter, water and ash are determined. All these attempts have not given satisfactory results owing to the varied composition of the volatile matter in various coals.

Air Required for Combustion and the Products of Combustion. — As already pointed out, the combustion of 1 lb. of C to CO_2 requires $\frac{2 \times 16}{12} = 2.66$ lb. O, and that of 1 lb. of H to H_2O requires $\frac{.5 \times 16}{1.0} = 8$ lb. O. Since air contains in each pound only .23 lb. of oxygen, the air required for the two cases will be $\frac{2.66}{.23} = 11.57$ lb. and $\frac{8}{.23} = 34.78$ lb. respectively.

The products of combustion are, in the case of carbon, 3.66 pounds of CO_2 and, in the case of hydrogen, 9 pounds of H_2O, if the theoretical amount of oxygen has been used in each case.

With the aid of these fundamental figures, the air required and the resulting products of combustion may be computed for any given fuel, solid, liquid or gaseous, if the composition of the fuel is known.

In general, any fuel which contains per pound say C lb. carbon, H lb. hydrogen and O lb. oxygen, requires

$$\frac{2.66\,C + 8\,H - O}{.23} \text{ pounds of air for its complete combustion.}$$

In gas-engine practice, however, the case of gas fuels is of much more importance, and for this we can derive the following general formulæ:

(*a*) For One Pound of Gas. — Let the gas consist of

N_1 lb. CO + N_2 lb. H + N_3 lb. CH_4 + N_4 lb. C_2H_4 + N_5 lb. C_2H_2 + N_6 lb. O + N_7 lb. N + N_8 lb. CO_2 + N_9 lb. H_2O = 1 pound.

Air theoretically required per pound of gas

$$= \frac{.57\ N_1 + 8\ N_2 + 3.99\ N_3 + 3.43\ N_4 + 3.07\ N_5 - N_6}{.23} \text{ pounds} \tag{6}$$

Products of combustion per pound of gas

$$CO_2 = [1.57\ N_1 + 2.74\ N_3 + 3.14\ N_4 + 3.38\ N_5 + N_8] \text{ pounds} \tag{7}$$

and

$$H_2O = [9\ N_2 + 2.25\ N_3 + 1.29\ N_4 + .69\ N_5 + N_9] \text{ pounds} \tag{8}$$

Besides the above amounts of CO_2 and H_2O, the products of combustion will also contain any excess oxygen that may have been used together with the nitrogen brought in by the oxygen.

If the analysis of the exhaust gas shows free oxygen, the excess coefficient for the air used, provided the fuel gas itself carries no nitrogen, may be computed from the formula

$$U = \frac{N}{N - 3.76\ O}$$

where N = per cent of nitrogen, O = per cent of free oxygen in the exhaust gas, by volume and 3.76 is the volume ratio of N to O in air. This formula, together with the two given above for CO_2 and H_2O, allow of a complete determination of the products of combustion.

In case the fuel gas itself carries N, the determination of the exess coefficient for the air used is not so simple and should be made according to the method outlined on p. 144.

(*b*) For One Cubic Foot of Gas. — Assume that the gas has the following analysis:

N_1 per cent by volume of CO
N_2 per cent by volume of H_2
N_3 per cent by volume of CH_4
N_4 per cent by volume of C_2H_4
N_5 per cent by volume of C_2H_2

N_6 per cent by volume of O_2
N_7 per cent by volume of N_2
N_8 per cent by volume of CO_2
N_9 per cent by volume of H_2O

Air required per cubic foot, theoretically,

$$= \frac{\frac{N_1 + N_2}{2} + 2\,N_3 + 3\,N_4 + 2.5\,N_5 - N_6}{.21} \text{ cubic feet} \qquad (9)$$

Products of combustion per cubic foot of gas

$$CO_2 = [N_1 + N_3 + 2N_4 + 2N_5 + N_8] \text{ cubic feet} \qquad (10)$$

$$H_2O = [N_2 + 2N_3 + 2N_4 + N_5 + N_9] \text{ cubic feet} \qquad (11)$$

Besides these amounts of CO_2 and H_2O there will in most cases be additional amounts of free oxygen and of nitrogen. The volumes of these can be determined from the exhaust gas analysis as before.

The table, page 139, gives the main constants for the principal gases met in gas engine practice.

The constant, R, can be computed from

$$\frac{R}{J} = (C_p - C_v) = \frac{2}{m}$$

where,

$J =$ the mechanical equivalent of heat $= 778$

and

$m =$ molecular weight of the gas.

Sample Computation. — Given the following chemical analysis of a producer gas, p. 140, to determine its heating value, its molecular weight m, C_p and C_v, R and $\frac{C_p}{C_v}$. Also these quantities for various mixtures of this gas with air, and the amount of and the constants for the burned gases after combustion of these fuel mixtures.

Gas		Lower Heating Value per		Weight per cu. ft. lbs. at 29.92 inches 32° F.	Theoretical Amount of			Products of Combustion		Specific Heat * per pound		Constant R in equation
		Lb.	Cu. ft.		Oxygen for 1 lb. lbs.	Air for 1 lb. lbs.	Air for 1 cu. ft. cu. ft.	CO_2 lbs.	H_2O lbs.	C_p	C_v	$\frac{pv}{T}=R$
Hydrogen	H_2	52230	297	.00559	8.00	34.78	2.40	—	9	3.409	2.412	775.6
Carbon Monoxide ..	CO	4380	342	.07807	.57	2.48	2.38	1.57	—	.245	.174	55.23
Methane...........	CH_4	21385	952	.04464	3.99	17.30	9.56	2.74	2.25	.593	.468	97.25
Ethylene	C_2H_4	20023	1564	.07809	3.43	14.95	14.58	3.14	1.29	.404	.333	55.23
Acetylene	C_2H_2	20673	1499	.07251	3.07	13.35	11.99	3.38	.69	.346	.270	59.13
Butylene	C_4H_8	19512	3040	.15590	3.43	14.95	28.86	3.14	1.29	.404	.333	
Nitrogen	N_2	—	—	.07831	—	—	—	—	—	.244	.173	55.23
Oxygen	O_2	—	—	.08921	—	—	—	—	—	.217	.153	49.79
Air		—	—	.08072	—	—	—	—	—	.2375	.1684	53.75
Water Vapor.......	H_2O	—	—	.05016	—	—	—	—	—	.48	.37	85.58
Carbon dioxide	CO_2	—	—	.12267	—	—	—	—	—	.20	.155	35.01

* From Güldner, p. 578.

SAMPLE OF DOWSON GAS, CLERK, p. 363

Components of Gas	Composition by Volume V	Composition by Weight W	Density Wt. per Cu. ft. Standard lbs.	Constants for Constituent Gases: Lower Heat Value per lb. B. T. U.	Molecular Weight m	C_v	C_p	R	$V \cdot m$	$W \cdot C_p$	$W \cdot C_v$	$W \cdot R$	Heating Value of Component Gases B. T. U.
H	.1873	.0153	.00559	52230	2	2.412	3.409	775.6	.3746	.0522	.0369	11.87	799.1
CO	.2507	.2902	.07807	4380	28	.174	.245	55.23	7.0196	.0711	.0505	16.03	1271.1
CH_4	.0031	.0020	.04464	21385	16	.468	.593	97.25	.0496	.0012	.0009	.19	42.8
C_2H_4	.0031	.0035	.07809	20025	28	.333	.404	55.23	.0868	.0014	.0011	.19	70.1
N	.4898	.5689	.07831	——	28	.173	.244	55.23	13.7144	.1388	.0984	31.42	——
O	.0003	.0005	.08921	——	32	.153	.217	49.79	.0096	.0001	.0001	.02	——
CO_2	.0657	.1196	.12267	——	44	.155	.200	35.01	2.8888	.0239	.0185	4.19	——
									24.14	.2887	.2064	63.91	2183.1
								=	m	C_p	C_v	R	B.T.U. per lb. of gas

Weight of gas per standard cubic ft. * $= \frac{m}{359} = \frac{24.14}{359} = .0673$ lbs.

Heating value per cu. ft. of gas = 2183.1 × .0678 = 146.9 B. T. U.

A briefer way of determining R would have been $R = J\ (c_p - c_v) = 778\ (.2887 - .2064) = 64.0$

* See page 127.

According to equation (9), the theoretical amount of air required by this gas per cubic foot

$$=\frac{\frac{.2507+.1873}{2}+(2\times .0031)+(3\times .0031)-.0003}{.21}$$

$$=\frac{.2342}{.21}=1.12 \text{ cubic feet.}$$

The following table gives in the first column the constants for the theoretical air-gas mixture. The second and third columns assume that an excess of air is used, in the first case equal to $1.5-1.12=.38$ cubic feet, in the last case equal to $2.0-1.12=.88$ cubic feet:

Constants for Various Mixtures of above Dowson Gas with Air

Ratio air to gas:			
By volume, V =	1.12	1.5	2
By weight, W =	1.35	1.79	2.39
Weight of standard cu. ft. of mixture, lbs. $=\frac{.0673+.08072\,V}{1+V}$	.0744	.0753	.0772
Heating value of standard cu. ft. of mixture, B. T. U. $=\frac{146.9}{1+V}$	69.4	58.8	49.0
$R=\frac{64+53.7\,W}{W+1}$	58.2	57.6	56.9
$C_p=\frac{.2887+.238\,W}{W+1}$	.2595	.2560	.2529
$C_v=\frac{.2064+.169\,W}{W+1}$	.1849	.1824	.1800
$\frac{C_p}{C_v}=$	1.403	1.404	1.405

The next table gives the corresponding constants for the burned gases resulting from the combustion of the fuel mixtures assumed above.

The changes occurring during combustion cause a change in the values of R, C_p, C_v, $\frac{C_p}{C_v}$ etc. For the theoretical case, *i.e.*, with 1.12 cubic feet of air to 1 cubic foot of gas, the products of combustion will be according to equations (10) and (11).

CO_2 = (.2507 + .0031 + .0062 + .0657) = .3257 cubic feet.
H_2O = (.1873 + .0062 + .0062) = .1997 cubic feet.

Since 1.12 cubic feet of air were used, there must also necessarily be nitrogen to the volume of

$$[(1.12 \times .79) + .4898] = 1.3746 \text{ cubic feet,}$$

.4898 cubic feet being due to the nitrogen in the fuel gas itself.

The volumes of the exhaust gases in the second column of the table are found as follows. The volumes of CO_2 and of H_2O are of course the same as before, since 1 cubic foot of gas is burned in every case. But since only 1.12 cubic feet of air are required and 1.5 cubic feet have been used, the excess air is 1.5 − 1.12 = .38 cubic feet. This consists of .38 × .79 = .3002 cubic feet of N and .0798 of O. Hence the excess O appearing will be .0798 cubic feet, while the N now is 1.3746 + .3002 = 1.6748 cubic feet.

Constants for the Burned Gases

Ratio air to gas by vol.	1.12	1.5	2.0
Vol. of exhaust gases, cu. ft. CO_2	.3257	.3257	.3257
H_2O	.1997	.1997	.1997
O	.0000	.0798	.1848
N	1.3746	1.6748	2.0698
Vol. of exhaust gases to 1 cu. ft. of Dowson gas, cu. ft. V_2	1.9000	2.2800	2.7800
Vol. of mixture before Combustion V_1	2.12	2.50	3.00
Ratio $\frac{V_2}{V_1}$	.896	.912	.926
% contraction	10.4	8.8	7.4
Principal Burned Gas Constants: $R_e = R\frac{V_2}{V_1}$	52.1	52.5	52.7
C_p per pound	.2478	.2462	.2446
C_v per pound	.1809	.1787	.1769
$\frac{C_p}{C_v}$	1.369	1.379	1.383

The values of C_p and C_v in the foregoing table are found as follows. Consider the mixture with ratio = 1.5:

Burned Gas	By Volume	By Weight
CO_2	.3257	.0399
H_2O	.1997	.0100
O	.0798	.0071
N	1.6748	.1311
	2.2800 cu. ft.	.1881 pounds

$$
\begin{array}{llll}
\text{For } CO_2 & C_p = .0399 \times .20 & = .0080 \\
\quad H_2O & C_p = .0100 \times .48 & = .0048 \\
\quad O & C_p = .0071 \times .217 & = .0015 \\
\quad N & C_p = .1311 \times .244 & = .0320 \\
\quad \text{For} & .1881 \text{ lbs. } \Sigma C_p & = .0463
\end{array}
$$

$$C_p = \frac{.0463}{.1881} = .2462$$

Since $C_p - C_v = \frac{R}{J}$, $C_v = C_p - \frac{R}{J}$

$$C_v = .2462 - \frac{52.5}{778}$$

$$= .1787$$

$$\text{and } \frac{C_p}{C_v} = 1.379$$

Attention should be called to the fact that although considerable contraction of volume occurs, in the case of this gas, during combustion, still the values of R, C_p and C_v are not greatly different from the corresponding values before combustion. In some other gases, as illuminating gas for instance, the change is even less. So that in most ordinary cases it is sufficiently accurate to assume that these gas constants are the same before and after combustion. Only in cases where extreme accuracy is desired is this assumption not permissible.

3. Computation of the amount of air used in excess of theoretical requirements from the exhaust gas analysis.

In actual practice the exhaust gases are analyzed for CO_2, O, and N. By the ordinary method of collecting these gases, the water vapor originally present is thrown down and does not appear in the analysis. As mentioned before, if the fuel gas itself carries

no nitrogen, the excess coefficient for the air actually used may be computed from the formula given on page 137.

To show an example of the method of computation when the fuel gas carries N, we will take the case of the Dowson gas above given, and assume that the exhaust gas analysis gives the following results: CO_2 – 14.36%, O – 5.31%, and N – 80.33% by volume. We proceed as follows:

Products of Combustion for theoretical ratio per cubic foot of gas, are:

CO_2	.3257 cu. ft.		
H_2O	.1997 "		
O	.0000 "		
N	.4898 "	due to gas itself	= 1.3746 cu. ft. of N, of which 35.63% is due to gas. 64.37% is due to air.
N	.8848 "	due to air used	
Total	1.9000 cu. ft.		

On the basis of the above exhaust gas analysis, we now have:

Total N 80.33
Of this amount, N due to excess air will be 3.76×5.31 . . = 19.96
Leaves N due to the gas itself and to air actually burned = 60.37
Of this remainder, as above shown, 35.63% is due to the fuel gas . . . . = 21.49
Leaves N due to the air actually burned . . . . = 38.88

Hence the excess coefficient

$$U = \frac{38.88 + 19.96}{38.88} = \frac{58.84}{38.88} = 1.5$$

and the real ratio of air to gas for the original fuel mixture was $1.5 \times 1.12 = 1.68$.

4. Calorific Intensity. — By calorific intensity is meant the temperature that can be realized theoretically when a unit weight of any fuel is completely burned under stated conditions of oxygen or air supply. If H represents the heating value of the fuel in B. T. U., A, B, C, etc., the weights of the various resulting products of combustion, and C_{pA}, C_{pB}, C_{pC}, etc., the specific heat at constant pressure of these products, the general statement for calorific intensity, supposing the pressure to remain constant, is

$$\text{Theoretical Temperature Rise} = \frac{H}{AC_{pA} + BC_{pB} + CC_{pC} +} \,^\circ F.$$

Thus the calorific intensity of hydrogen with theoretical air would be, the products of combustion being water vapor and nitrogen,

$$\frac{52230}{[9 \times .48] + [26.8 \times .244]} = 4800^\circ F.$$

That of C to CO_2 with theoretical air would similarly be

$$\frac{14500}{[3.66 \times .20] + [8.91 \times .244]} = 5000^\circ F.$$

Such high temperatures, however, are practically never realized, due probably to two causes. On the one hand it is claimed that the specific heat of gases is not constant at all temperatures, but that it rises with the temperatures; on the other, dissociation is supposed to set in before such temperatures are reached. These matters will be taken up somewhat more in detail in a later chapter.

CHAPTER VII

GAS-ENGINE FUELS; THE SOLID FUELS; GAS PRODUCERS

THE general requirement for a gas-engine fuel is that it must mix readily with air to form a combustible gas or vapor. Further, it should burn with little or no residue. This latter requirement is not met by the solid fuels, as coal dust for instance, and while isolated attempts at using powdered coal directly have been made, they have so far not been successful, owing to the fact that the resulting ash soon seriously interferes with operation.

The gas-engine fuels may be classed under three heads:

1. **The solid fuels.**
2. **The liquid fuels.**
3. **The gas fuels.**

It is the rule that the working medium in all internal combustion engines is either a combustible gas or a combustible vapor, no matter what the fuel may have been from which it was derived. This implies gasification of the solid, and vaporization of the liquid, fuels.

As already pointed out above, the solid fuels cannot be employed in their natural state. From coal we derive by distillation illuminating gas, and from coal and sometimes other materials, as wood, refuse, etc., by gasification, the various classes of producer or power gas.

Illuminating gas will be further considered under the head of gas fuels.

1. The Conversion of the Solid Fuels to Gas: Producer Gases. — Gasification of solid fuel differs from distillation in the fact that the process is carried one step further, *i.e.*, not only are the gases, if any, driven off from the fuel, but the carbon itself is gasified, leaving behind nothing but ash.

The fundamental principle of all producer-gas processes is

therefore, first, dry distillation of the fuel, and, second, the conversion of the solid carbon into a combustible gas, which can only be carbon monoxide. If the producer gas is found to contain other gases than those mentioned, it can only be due to changes in the process, unavoidable or otherwise.

Producer practice may be carried on in the following ways:

1. No steam or water introduced with the air, resulting in air gas.

2. Producer blown up with air for one period, then blown with steam alone. Product during first stage is air gas, during the second, water gas.

3. Producer furnished with air carrying a certain quantity of water vapor. Product is ordinary producer gas, Dowson gas, etc.

The first of these is seldom employed on account of limitations pointed out below. The second and third introduce modifications into the simple air-gas process owing to the presence of water or steam.

Air Gas. — Considering the case of the gasification of carbon alone, resulting in the production of the so-called air gas, assume the combustion of C to CO complete; we then have

$$1 \text{ lb. of C} + 1.33 \text{ lb. O} = 2.33 \text{ lb. CO}.$$

If C had been burned completely to CO_2, the calorific power would have been 14647 B. T. U.; burning only to CO, however, we obtain only 4429 B. T. U., so that the remainder or 10218 B. T. U. is carried out of the producer by the gas made, and thus represents its heat energy. 4429 B. T. U. appears as sensible heat in the gas, and if the gas is cooled before entering the engine cylinder, as it usually is for good reasons, the greatest possible efficiency which can be realized from the gasification of 1 pound of carbon in this way is $\frac{10218}{14647} = 69.6$ per cent. It will be shown below that this is by no means the maximum possible producer efficiency.

Water Gas. — When water vapor is led through or over incandescent carbon the following reactions take place:

$$\text{I. } C_2 + 4\,H_2O = 2\,CO_2 + 4\,H_2.$$

$$\text{II. } C_2 + 2\,H_2O = 2\,CO + 2\,H_2.$$

I occurs at temperatures less than 1250 degrees Fahrenheit, while II alone occurs at temperatures exceeding 1800 degrees Fahrenheit; both may occur between these temperature limits, but the higher the temperature the greater the formation of CO. The maximum amount of CO is of course the end in view, and assuming that no CO_2 is formed, *i.e.*, temperature at or above 1800 degrees Fahrenheit, we have the following quantitative statement:

$$1 \text{ lb. C} + 1.5 \text{ lb. } H_2O = 2.33 \text{ lb. CO} + .17 \text{ lb. } H_2$$

from which 1 pound of water gas must contain

$$\frac{2.33}{2.33 + .17} = .932 \text{ lb. Carbon monoxide}$$

and

$$\frac{.17}{2.33 + .17} = .068 \text{ lb. Hydrogen}$$

The gasification of 1 pound of carbon, therefore, in the presence of water vapor results in products which, on complete combustion, develop the following amount of heat:

2.33 lbs. CO × 4380 =	10205 B. T. U.
.17 lbs. H_2 × 62100 =	10557 B. T. U.
Total,	20762 B. T. U.
Heating value of C to CO_2,	14647 B. T. U.
Excess,	6115 B. T. U.

The excess of 6115 B. T. U. can only be due to heat rendered latent during the process. Water vapor on coming in contact with incandescent carbon dissociates into H_2 and O. The latter unites with C to form CO_2, but as the temperature of the producer is at or over 1800 degrees Fahrenheit, CO_2 is dissociated to CO. The heat thus rendered latent accounts for the excess above shown. Now it is evident that this heat can only come from the stock of heat present in the producer when the blowing with steam first starts. Hence there must be a continual cooling of the producer contents during the period of water-gas making. This results finally in a serious production of CO_2 according to reaction I, when the steam must be shut off and the contents of the producer brought back to incandescence by blowing with air.

The most unfavorable operation of the producer occurs when the reactions are according to equation I. Under this condition

$$1 \text{ lb. C} + 3 \text{ lb. } H_2O = 3.66\ CO_2 + .34\ H_2$$

so that 1 pound of water gas then contains

$$\frac{3\ 66}{3.66 + .34} = .915 \text{ lb. } CO_2$$

and

$$\frac{.34}{3.66 + .34} = .085 \text{ lb. Hydrogen}$$

The production of water gas reckoned on the basis of coal or carbon is not at all efficient, since the heat in the poor gas made during the blowing-up period is very often wasted, and only in rare instances of utility. Other losses are, of course, those through incomplete combustion, radiation, etc.; but these are inherent in all producers to a greater or lesser extent.

Producer Gas. — Midway between air gas and water gas we find the great class of power gases for the production of which the producer is blown continuously with a mixture of air and water vapor.

To get a fair insight into the working of a power gas producer and of the efficiencies that may be realized, we will assume the following definitions and quantities.*

1. The heat supplied to a producer consists of the heat furnished to it in the fuel plus the heat contained in steam and air above a certain fixed temperature, say 32 degrees Fahrenheit.

2. The heat leaving the producer in the gas is made up of the latent heat of the gas plus the sensible heat. What the quantity of heat considered as the useful effect in efficiency should be depends upon circumstances. In furnace work, where it may be of advantage to employ the hot gas, the useful effect would be the sum of the latent and sensible heats of the gas. In gas-engine practice, on the contrary, the opposite is the case, and the useful effect would be the latent heat only. In the first case we speak of the *hot-gas efficiency*, in the second of the *cold-gas efficiency*.

* Adapted from the discussion of E. Meyer, Zeitschrift des Vereins deutscher Ingenieure, 1895, p. 1523.

3. Inside of the producer the following reactions take place, some endothermic, others exothermic.

Of every pound of carbon the larger part burns to CO, the remainder to CO_2. The heat generated from these two combustions is utilized in the following ways: Part of it dissociates the steam present, forming H, and CO or CO_2, or both. How this action varies with the temperature of the producer has already been pointed out. A second part of the heat serves to bring the fresh fuel up to the temperature of the producer, a third is lost by radiation from the exterior producer walls, and the remainder appears as sensible heat in the gas made.

The formulæ to be derived will be based upon one pound of C rather than upon one pound of coal, for the reason that coals vary greatly in composition, and it is in every case quite easy to change from this basis to that of coal if the qualities of the coal be known. Let

14500 B. T. U. = heat of combustion of 1 lb. of C to 3.66 lb. CO_2.
4400 B. T. U. = heat of combustion of 1 lb. C to 2.33 lbs. CO.
10100 B. T. U. = heat of combustion of 2.33 lb. CO to 3.66 lb. CO_2
6900 B. T. U. = $\frac{62100}{9}$ = heat required to dissociate 1 lb. of water vapor under producer conditions.

x = part of 1 lb. of C burning to CO_2.
$(1 - x)$ = part of 1 lb. of C burning to CO.
y = pounds of steam introduced per lb. of C.
A = heat furnished in steam.
B = heat furnished in air.
C = heat required to bring fresh fuel to temperature of producer.
R = heat lost by radiation.
S = sensible heat of the gas.

All of the above heat quantities are per pound of carbon gasified, and above a temperature of say 32 degrees Fahrenheit.

With this notation the general heat equation for the producer may be stated as follows, based on 1 lb. of carbon:

$$4400\,(1 - x) + 14500\,x + A + B = 6900\,y + S + R + C$$

The heat that will be generated by the combustion of the volume of gas formed comes from CO and H.

Heat generated in gas per pound of C gasified

$$= [10100\ (1 - x) + 6900\ y.]\ \text{B. T. U.}$$

6900 y for the heat generated by H is obtained by considering that we must receive as much heat from the combustion of the H in the gas as was rendered latent during dissociation of the amount of H_2O required to furnish it.

Heat supplied per pound of C gasified = 14500 + A + B. Hence

$$\text{Cold-gas efficiency} = \frac{10100\ (1 - x) + 6900\ y}{14500 + A + B}$$

$$\text{Hot-gas efficiency} = \frac{10100\ (1 - x) + 6900\ y + S}{14500 + A + B}$$

In the latter case, no part of the sensible heat of the gas is abstracted before the gas is used. In practice there is always some loss of temperature between the producer and the place where the gas is used, hence S is never fully obtained.

If in the above general heat equation the values of x, A, B, C, R and S are known, the value of y, *i.e.*, the pounds of steam to be used per pound of carbon, may be computed. The composition of the resulting producer gas may be computed as follows, using the above notation:

COMPOSITION BY WEIGHT PER POUND OF CARBON GASIFIED.

$$CO = (1 - x)\frac{28}{12} = 2.33\ (1 - x) \text{ pounds}$$

$$CO_2 = \frac{44}{12}x = 3.66\ x \text{ pounds}$$

$$H = \frac{2}{18}y = \frac{y}{9} \text{ pounds}$$

The amount of N is found as follows:

$$\text{Oxygen required for } CO = (1 - x)\ \frac{16}{12} = 1.33\ (1 - x) \text{ pounds}$$

$$\text{Oxygen required for } CO_2 = \frac{32}{12}x = 2.66\ x \text{ pounds}$$

Oxygen produced by dissociation of H_2O.

$$= \frac{16}{18}y = \frac{8}{9}y \text{ pounds}$$

$\therefore$ O required from air blast

$$= 1.33\,(1 - x) + 2.66\,x - \frac{8}{9}\,y \text{ pounds}$$

$$= 1.33\,(1 + x) - \frac{8}{9}\,y \text{ pounds}$$

Hence air required $= \left[1.33\,(1 + x) - \frac{8}{9}\,y\right] \frac{100}{23.5}$ pounds

Hence also N brought in by air blast is

$$N = \left[1.33\,(1 + x) - \frac{8}{9}\,y\right] \frac{76.5}{23.5} \text{ pounds}$$

The total weight of gas produced by one pound of carbon therefore is

$$= 2.33\,(1 - x) + 3.66\,x + \frac{y}{9} + \left[1.33\,(1 + x) - \frac{8}{9}\,y\right] \frac{76.5}{23.5} \text{ pounds}$$

$$= (6.67 + 5.67\,x - 2.77\,y) \text{ pounds of producer gas.}$$

COMPOSITION BY VOLUME PER POUND OF CARBON GASIFIED.

Volumes at 32 degrees and 14.7 lb. pressure.

Weight per cubic foot of the various gas under standard conditions.

$CO = .07807$ lb., $CO_2 = .12267$ lb., $H = .00559$, $N = .07831$.

Hence, from the above weight computations, we may write directly.

$$\text{Volume of CO} = \frac{2.33\,x}{.07807}\,(1 - x) = 29.84\,(1 - x) \text{ cu. ft.}$$

$$\text{Volume of } CO_2 = \frac{3.66}{.12267}\,x = 29.02\,x \text{ cu. ft.}$$

$$\text{Volume of H} = \frac{y}{9 \times .00559} = 19.87\,y \text{ cu. ft.}$$

$$\text{Volume of N} = \left[1.33\,(1 - x) - \frac{8}{9}\,y\right] \frac{76.5}{23.5 \times .07831}$$

$$= (55.51 + 41.74\,x - 37.10\,y) \text{ cubic feet.}$$

From this

Total volume of gas per pound of C gasified

$$= [29.84(1 - x) + 29.02\,x + 19.87\,y + 55.51 + 41.74\,x - 37.10\,y] \text{ cubic feet.}$$

$$= (85.35 + 40.92\,x - 17.23\,y) \text{ cubic feet.}$$

2. Theoretical Yield of Producer. — If it is supposed that no CO_2 is formed, that all the sensible heat produced in the generator is recovered in making steam and preheating air and fresh fuel, and that no radiation has taken place, we shall have

$$x \text{ and } R = 0, \text{ and } S = A + B.$$

Under these theoretical conditions, the general heat equation (p. 151) then becomes

$$4400 = 6900\,y + C.$$

The value of C is approximately $.2 \times 1800 = 360$ B. T. U. Hence

$$4140 = 6900\,y,$$

from which $y = \frac{4140}{6900} = .600$ pounds of steam per pound of C gasified.

The theoretical yield of gas per pound of carbon under these conditions will be

29.84 cubic feet of CO.

$19.87 \times .600 = 11.92$ cubic feet of H

and

$55.51 - 37.10 \times .600 = 33.25$ cubic feet of N.

Total volume of yield = 75.01 cubic feet per pound of C.

Composition of gas, per cent by volume,

39.8 per cent CO, 15.9 per cent H, 44.3 per cent N.

Taking the higher heating value of H at 346 B. T. U. per cubic foot, and the heating value of CO at 342 B. T. U. per cubic foot, the gas yield from 1 pound of C will develop

$$(29.84 \times 342) + (11.92 \times 346) = 14249 \text{ B. T. U.}$$

The higher heating value per cubic foot of this theoretical gas is therefore

$$\frac{14249}{75.01} = 187.2 \text{ B. T. U. per cu. ft.}$$

In actual practice, however, x and R cannot $= 0$, neither is $S = A + B$; *i.e.*, not all of the heat appearing as sensible heat in the gas leaving the producer is ever recovered. To

make clear what happens under these circumstances, the following table is constructed. In this table it is assumed that the heat furnished in steam and air per pound of carbon equals 1000 B. T. U. = $(A + B)$ in all cases, and that the sum of the heat losses due to radiation, R, and sensible heat of the gas, S, = 1640 B. T. U. per pound of carbon.

The heat furnished the producer per pound of C will then in all cases be 14500 + 1000 = 15500 B. T. U., while the heat accounted for will be 15500 − (1640 + 360) = 13500 B. T. U., so that the generator efficiency on hot gas in all cases = $\frac{13500}{15500}$ = 87.2 per cent. It is further assumed for illustration that only the amount of carbon burned to CO_2 per pound of carbon gasified varies.

% of Carbon Burned to CO_2	Steam used lbs.	Cubic feet of gas per pound of C Gasified						Composition, % by volume				Lower Heating Value per cu. ft. at 32° and atmo. press.
		CO_2	CO	H	N	Total		CO_2	CO	H	N	
						cu. ft.	lbs.					
0	.493	0	29.84	9.80	37.22	76.86	5.30	0	38.9	12.6	48.5	170.0
10	.639	2.90	26.85	12.70	35.94	78.39	5.47	3.7	34.2	16.1	46.0	164.7
20	.783	5.80	23.90	15.55	34.84	80.10	5.63	7.2	29.9	19.4	43.5	159.7
30	.932	8.70	20.90	18.50	33.51	81.60	5.79	10.7	25.6	22.7	41.0	155.0
40	1.079	11.60	17.90	21.45	32.18	83.13	5.94	13.6	21.5	25.8	39.1	150.2

In commenting upon the above table, Meyer points out that as the percentage of CO_2 in the gas increases, the heating value of the gas decreases, but since at the same time the percentage of H and the volume of gas per pound of C increase, it is not always right to conclude from analysis alone that the efficiency of the generator is less with a fairly high than with a low percentage of CO_2 in the gas. In general terms it can be stated, however, that the lower the temperature of the generator, the greater the formation of CO_2, and the greater also the loss of heat in sensible heat of the resulting gas.

Referring to the gas engine itself, a high percentage of CO_2 in the gas means a low engine capacity, since this high percentage is usually also accompanied by an increased amount of the other indifferent gases.

3. **Gas Producers in Practice.** – Turning now to actual generator practice, we find the following main points of difference: The fuel is not pure carbon, but some impure form of it, as coal, coke, lignite, peat, or wood. This in itself merely results in a lower yield of gas per pound of fuel fired than that above computed, and this decrease is further emphasized by the fact that some of the unburned carbon in the fuel is always lost in the ash. A complex fuel being used containing gases which are distilled off during the first part of the process, the resulting producer gas will have a somewhat different composition than that above computed. The main difference is due to the addition of hydrocarbons, and this difference is therefore greater with bituminous coals than with any of the other fuels. The use of any of the above-mentioned fuels results also in other complications more or less difficult, depending upon the fuel used. Such are the formation of tar, dust carried by the gas, etc., all of which make a cleaning of the gas imperative before it can be used.

The primary consideration in the operation of the gas producers is perhaps the kind of fuel used. The points to be considered in this connection are: percentage of water carried by fuel, amount and kind of ash, tar-forming ingredients of the fuel, size of fuel, and whether it cokes or not.

A high percentage of water has a direct effect in lowering the temperature of the producer, besides lowering the heating value

of the gas per cubic foot as made. A large amount of ash makes more frequent cleaning out necessary, or it is likely to result in a partial stoppage of the air supply. If the ash should be easily fusible the case is much more complicated, as this results in bad clinkering. The size of the fuel should be a happy medium. Small fuel, *i.e.*, screenings, etc., clog up easily and in any case require a higher blast pressure. Large fuel, on the other hand, offers too little surface for gasification and is apt to let much water and CO_2 escape unreduced. A coking coal nearly always gives trouble from this cause, and it necessitates constant breaking up of the charge.

The formation of tar, which results especially when bituminous coals are gasified, makes a cleaning of the gas for engine purposes indispensable. Tar results when some of the hydrocarbon gases are condensed through cooling in the gas mains and pipes. If these gases reach the cyclinder their combustion is likely to result in a strong deposit of soot. In either case the operation of the engine will soon be seriously interfered with. Tar can be almost entirely removed from the gas by washing it, but this process requires constructions fully as costly as the producer itself and hence other methods have been employed.

The tar-forming gases are always those which are formed from the dry distillation of the coal, hence most trouble is encountered with bituminous coal, less with lignite and still less with anthracite. For this reason anthracite and coke producers have been most successful, although producers using brown coals and lignites are in operation, as are also those using bituminous coal, but with less success. This does not apply to steel works where bituminous coal is used extensively for gasification. But there the gas is used mostly hot and less trouble from tar is experienced.

The tar gases can be "fixed," *i.e.*, changed to permanent gases when the producer gas containing them is led through an incandescent bed of fuel before entering the gas mains. In this case the tarry hydrocarbons are changed either to H_2O and CO_2 or split up into CO and H. In some producers only the gases resulting from the dry distillation are handled in this way. In either case the tarry hydrocarbons are fixed, and no elaborate cleaning apparatus for the gas is required. The necessity for

treating bituminous producer gas in this way has resulted in various constructions of producer, a few of which are given below.

Gas producer installations may be divided into three classes:

a. Pressure Producers. — In these air and steam are furnished to the fuel bed by a blower or fan. The ash pit of the producer must be enclosed, making the removal of ash complicated or the action of the producer intermittent, unless the water-bottom type is used. Steam for blowing is usually furnished by a separate boiler. Since the rate of production of gas is usually not regulated according to the demand for gas directly, a gas holder is usually necessary for this type.

b. Suction Producers. — In this class the air and steam are drawn through the producer by the suction of the engine cylinder. The production of gas is thus directly regulated by the demand. The ash pit remains open, and steam enough can usually be generated by the sensible heat of the gas.

Suction gas producers have nearly replaced pressure producers for gas-engine purposes. Some of the obvious advantages, as open ash pit, absence of separate boiler and of gas holder, have been pointed out above. The dangers at first supposed to be inherent in this system have failed to materialize. Leaks in a pressure system may lead to a poisoning of the atmospheric air by the color- and odorless CO, positively dangerous to attendants. Leaks in a suction system only result in an in-leakage of air. That this can never happen to such an extent as to form an explosive mixture, except through a combination of extraordinary circumstances, is at once evident when we consider that the ratio of air to producer gas for such a mixture would have to be at least 1 to 1.

In spite of such advantages the suction system is by no means perfect. The regulation of the water supply to control the amount of H in the gas, the vaporizer for the water, and the cleaning apparatus are still points which admit of improvement even in the most recent form.

c. Combination Producers. — The air and steam mixture is drawn through the producer by a fan and the resulting gas forced by the same fan to the engine. The producer in this system is thus of the suction type.

Pressure Producers.—*Taylor.* Fig. 7-1 shows the Taylor producer made by R. D. Wood & Co. of Philadelphia. In their publications on the producer this company lays down the following requirements for a successful pressure producer. Most of these, however, apply to producers in general.

Fig. 7-1.—Taylor Producer.

1. A continuous and automatic feed; the former for regularity and uniformity of gas production with improved quality, the latter for eliminating negligence of attendants.
2. A deep fuel bed carried on a deep bed of ashes; the first to make good gas, and the second to prevent waste of fuel.
3. Blast carried by conduit through the ashes to the incandescent fuel.
4. Visibility of the ashes, and accessibility of the apertures for their removal, arranged so that operator can see what he is doing.
5. Level, grateless support for the burden, insuring uniform depth of fuel at all points, and consequent uniformity in the production of gas.

These points are well covered in the design of the producer. The fuel is admitted through a distributing hopper which keeps the layer of fuel level over the cross-section. The bed of ashes is kept at about 6 inches over the top of the air pipe, thus protecting it from direct heat. The entire charge in the producer is supported by a plate whose diameter is somewhat greater than that at the bosh. As necessity requires, this plate can be revolved and the ashes are scraped, or they fall off, into the closed ash pit, which is under blast pressure. The grinding action ensuing when the plate is revolved settles the contents of the producer and thus closes up any free air channels that may have been formed. Once a day the pit must be opened for the removal of the ash. Blast is supplied generally by a steam jet.

Morgan. Somewhat similar in design is the Morgan producer, Fig. 7-2. The main point of difference is in the removal of the ash. The fuel here is also admitted through a continuous automatic feeding device. The blast is controlled by a steam injector so designed as to maintain a proper proportion between

air and steam. The make of gas can be completely controlled by the adjustment of a $\frac{3}{4}$-inch steam valve.

Fig. 7-2. — Morgan Producer.

This producer is of the water-bottom type, *i.e.*, the ashes fall into a water seal at the bottom, and may there be removed without stopping the operation of the producer. This is not easily done when a grate or similar device is used in a pressure producer. For certain fuels, especially those highly bituminous or those where ash is apt to clinker, the water-bottom producers possess some advantage over the others. About three feet above the water level in the ash pan, and some inches above the top of the blast distributing pipe, a number of sight holes are arranged around the circumference of the producer. Through these the zone of combustion may be watched. The top of the producer is covered with a shallow water pan. Poke holes through the top with water-sealed covers are also provided.

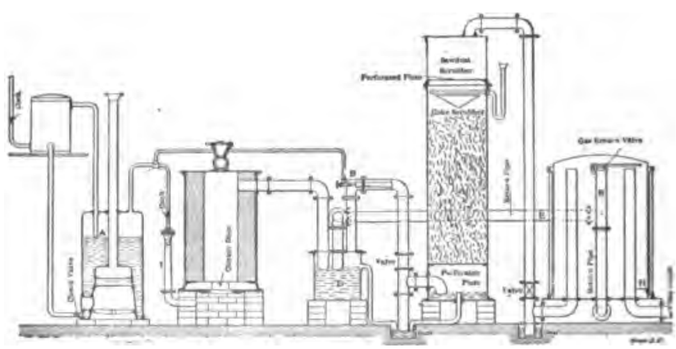


Fig. 7-3. — Wile Producer Installation.

Wile. A complete Wile pressure gas plant is shown in Fig. 7-3.* Steam under about 40 pounds pressure is generated in the boiler, *A*, and enters the generator through the injector, *I*, mixed with air. The gas made passes through the seal box, *D*, and the

* J. I. Wile in Power.

scrubber to the gas holder. This is the ordinary arrangement, but it is open to objection when the load is extremely variable. In such a case it is usual to arrange the gas holder so that it shuts off steam at *I* when the holder is full. The contents of the producer then cool, and a further cooling results when the steam is next turned on. The temperature ranges in the producer are therefore apt to be high under such conditions. To meet this difficulty the design can be changed by placing the steam injector at *B*, above the seal box *D*. The gas holder is connected with the seal box by a return pipe *E*. When the gas holder is up, catch *H* in the gas holder opens the return valve, and the injector *B* merely draws on the gas holder, placing the generator temporarily out of commission. When the gas holder falls, the return valve closes, the injector draws on the generator, and gas is again made. In this design the arrangement at *I* is then merely a saturator, and the plant is really a combination plant, the generator being under suction.

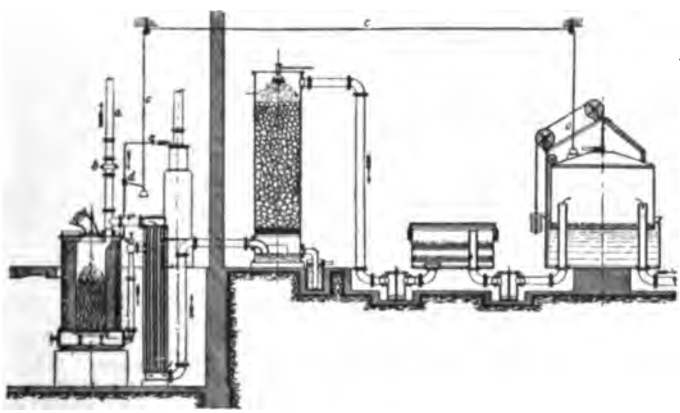


FIG. 7-4.—Koerting Pressure Producer.

*Koerting, Hannover.** Fig. 7-4 shows a Koerting pressure plant. The gases made during the firing-up period escape through the pipe *a*. Should the natural draft not be strong enough, it may be increased by means of a small steam blower in pipe *a*. After the flame at the try cock shows dark red, and not blue, the valve *b* is closed, and the gas made sent through the pre-heater, scrub-

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 384.

ber, and sawdust purifier to the gas holder. The make of gas is controlled by the gas holder through the chain *C*, which acts upon the blast through a throttle valve at *d*. According to published figures, the average analysis of the gas made is, by volume,

H, 18 per cent; CO, 26 per cent; $C_n H_{2n}$, 2 per cent.

CO_2, 7 per cent; N, 47 per cent; Efficiency, 80–82 per cent.

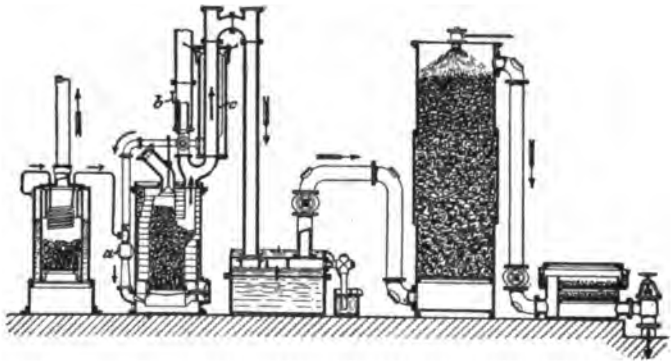


FIG. 7–5. — Deutz Pressure Producer.

Deutz.* The steam used for the blower *a*, in the Deutz plant, Fig. 7–5, is superheated in a coiled pipe above the fuel in the boiler.

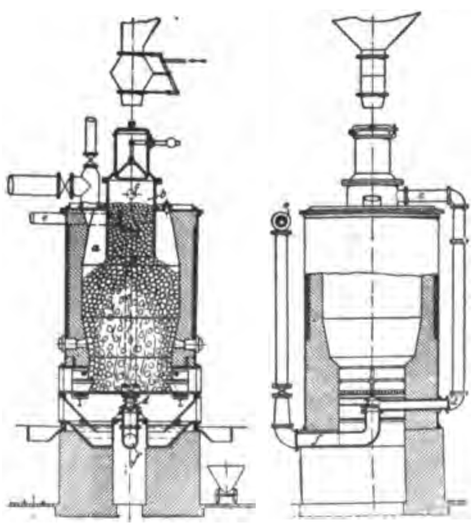


FIG. 7–6. — Poetter Producer.

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 384.

The air is pre-heated by the heat of gases in the heater *C*. The opening *b* is used to try the gas during firing up. Arrangements of scrubber and purifier are similar to those already described.

Poetter. The Poetter producer,* Fig. 7–6, is especially designed for bituminous coal. The fuel charged is distilled while still in the hopper, the gases formed are drawn off by special steam blower and are led through the pipe *Cd* under the grate. Air for blast is provided through *e f*. The gas made escapes at *a*, and is first used to raise the steam required in a boiler. It is then led through cooler, scrubber, coke- and sawdust purifier to the gas holder. Schöttler, in describing a plant of Poetter producers at Johannesburg, surmises that their operation might cause trouble, although the kind of coal used at Johannesburg is not stated.

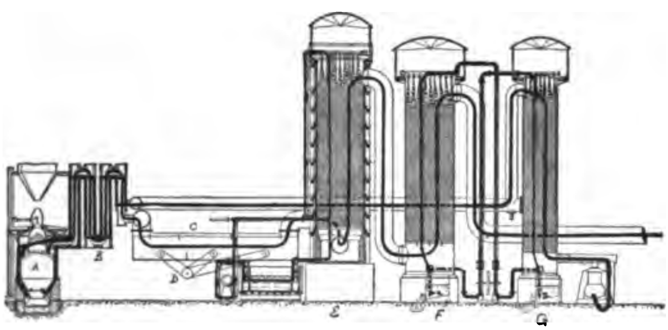


FIG. 7–7. — Mond Producer Plant.

Mond. When a producer is blown with a large excess of steam, a great deal of it will go through undecomposed. At the same time, however, the quality of the gas made undergoes some radical changes. The quantity of H in the gas will be high, sometimes up to 25 per cent, while on account of the low producer temperature, a great deal more of the C is burned to CO_2 than is ordinarily the case. A further, and important, change is that a great deal of the N is changed to ammonia, a reaction which does not take place in producers run under the ordinarily higher temperatures. To recover this ammonia is an important consideration. This is the process of Mond.

A Mond gas plant consists essentially of two parts, the producer and the condensing and recovery plant, Fig. 7–7.

* Rev. Mec., 1904, p. 484.

The details of the producer *A* do not differ much from those already described. It is of the water-bottom type. The gas made passes through the regenerators *B*. These consist of double wrought-iron tubes united alternately top and bottom. The hot gases pass through the inner tubes, heating the mixture of steam and air, used for blowing the producer, which flows through the outer tubes in the opposite direction. The gas next passes through the washer *C*. This is a large chamber partly filled with water. By means of dashers, driven by *D*, the chamber is kept filled with water spray. The gas is cooled considerably in passing through *C*, the water vapor in the gas being condensed to a great extent. Up to this point all of the apparatus is necessary even if no ammonia recovery is attempted.

For a gasification capacity of less than 30 tons of coal in twenty-four hours it is not usual to install a recovery plant on account of the high cost of installation. Beyond this capacity the installation is justified. A recovery plant consists of the acid tower *E*, the gas cooling tower *F*, and the air heating and saturating tower *G* (see Fig. 7–7).

The gas after leaving the condenser *C* enters the tower *E* at the bottom and flows upward through firebrick checker work. In doing so it meets a descending rain of sulfuric acid liquor containing about 4 per cent of free acid, and by this the free ammonia in the ascending gas is fixed, being changed to sulfate. The acid liquor is circulated by a special pump, and kept at the proper strength by drawing off the sulfate liquor and adding a corresponding amount of fresh acid solution from time to time.

The gas then enters the cooling tower *F*, at the bottom, and in its ascent is cooled by descending cold water. The gas gives up its burden of steam, which in turn heats the descending cold water. The gas is then conveyed to the gas mains.

The hot water leaving the gas-cooling tower is pumped to the top of the air-saturating tower. Here it flows downward through checker work, and in its descent saturates and heats the air, which is driven upward through the tower by blowers. Leaving this tower, the air-water vapor mixture then goes to the regenerator *B*, where it is further pre-heated before entering the producer.

Sexton* states that the amount of steam used is about 2½ tons per ton of fuel, and estimates that about two tons of this go through undecomposed.

The method has the advantage that slack coal may be gasified with success. The recovery of ammonia amounts to about 1 ton for 23 tons of coal gasified, or adding the fuel required for making steam, about 1 ton for 28.5 tons of fuel.

The average analysis of Mond gas is, by volume,

11 per cent CO, 17.1 per cent CO_2, 1.8 per cent CH_4, .4 per cent C_nH_{2n}, 27 per cent H, and 42.5 per cent N.

This gas is free from tar and excess of moisture, and burns with a non-luminous flame.

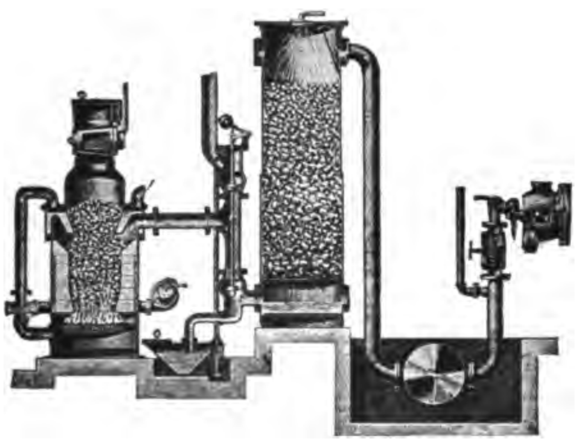


FIG. 7-8. — Deutz Suction Producer.

SUCTION PRODUCERS. — *Deutz.*† In the Deutz suction producer, Fig. 7-8, the vaporizer is arranged around the top of the generator. During the suction stroke of the engine, fresh air enters the vaporizer, is here mixed with water vapor, and then, flowing through the connecting pipe at the left side of the generator, reaches the under side of the grate. The gas made leaves the generator through the pipe at the right and enters the wet scrubber at the bottom. From here it passes to the engine through a

* Sexton, Producer Gas, p. 90.

† Güldner, p. 386.

regulator. Just before entering the engine the gas passes a vessel containing a wire brush for a final cleaning of tar. The larger Deutz suction installations show a somewhat different construction.

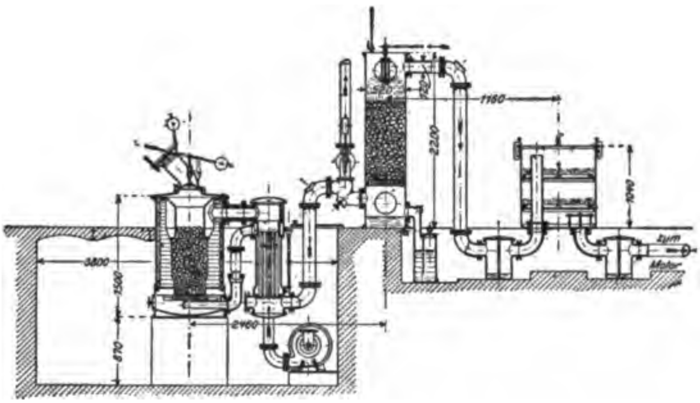


FIG. 7-9. — Koerting Suction Producer.

*Koerting.** Fig. 7-9 shows a Koerting suction plant designed for from 30-35 horse-power. In this case the vaporizer is independent of the generator. The latter is heated by the hot gases on their way to scrubber and purifier. During the heating-up period, the air is furnished by the blower shown. The smoke escapes through a pipe which branches off beyond the vaporizer, so that this will be heated during the blowing-up period.

FIG. 7-10. — Koerting Installation.

A complete view of a Koerting plant is shown in Fig. 7-10.

* Güldner, p. 386.

American Crossley. The American Crossley suction gas plants in this country are built by the Power & Mining Machinery Company. Fig. 7-11 shows a complete plant. The producer is of conventional design. The fuel bed rests on a shaking grate. The fuel is admitted as shown. The waste heat boiler or saturator surrounds the feed tube. The water level in this saturator is automatically regulated. During the firing-up period the producer is blown by means of the fan shown, the gases escaping by the purge pipe. A new idea seems to be the saturating of only part of the air supply. The makers point out the difficulty of

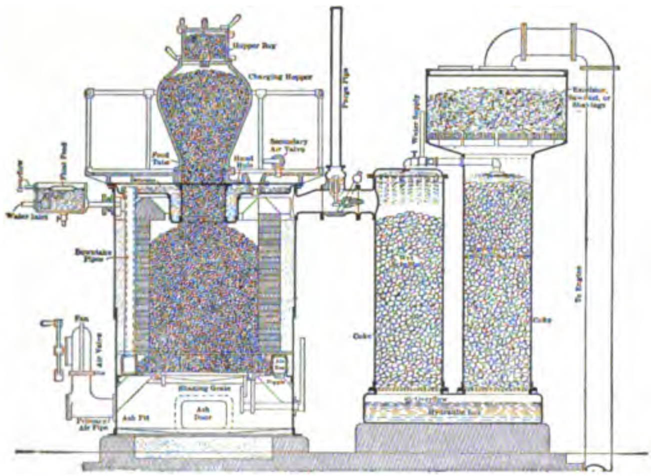


FIG. 7-11.—American Crossley Producer.

making uniform gas at all, especially at low, loads, owing to the difficulty of controlling the amount of steam admitted. To remedy this, part of the air only passes through the saturator, while the rest is directly admitted to the ash pit. Valves on these two air inlets can be set so as to give satisfactory gas over wide ranges of load.

Another arrangement in which this producer plant differs from most others is that of the cleaning apparatus. This consists of a wet scrubber, an hydraulic box, and a combination wet and dry scrubber. The gas enters the first and passes downward, passing upward in the combination scrubber. The operation is shown plainly in the cut.

Fairbanks-Morse. Fig. 7–12 shows a Fairbanks-Morse installation for 21 horse-power, the producer itself being shown in Fig. 7–13, in greater detail. The general design of the producer is very

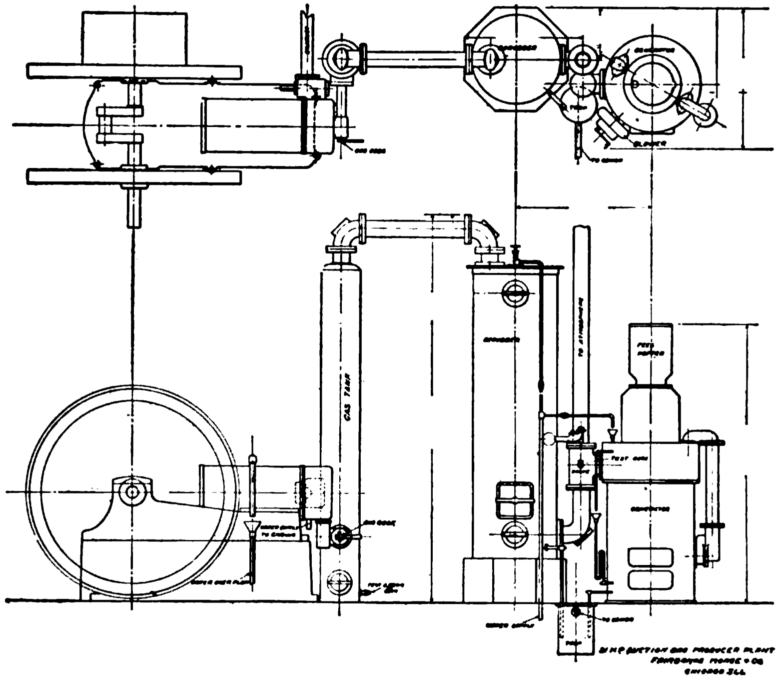


FIG. 7-12. — Fairbanks-Morse Installation.

similar to that of the American-Crossley, except that all of the air passes through the vaporizer. Apparently only a wet scrubber is provided, it being evidently intended to use anthracite only. A gas tank of considerable volume is interposed between scrubber and engine.

The suction producers so far described have been for anthracite, coke, or charcoal. The desire to utilize soft coal, lignites, etc., has led to special designs, of which the following are a few examples. In some of these all of the gas made from the fuel is passed through other highly incandescent but gas-free fuel to fix the tar-forming gases, in others only the gases of distillation are so treated.

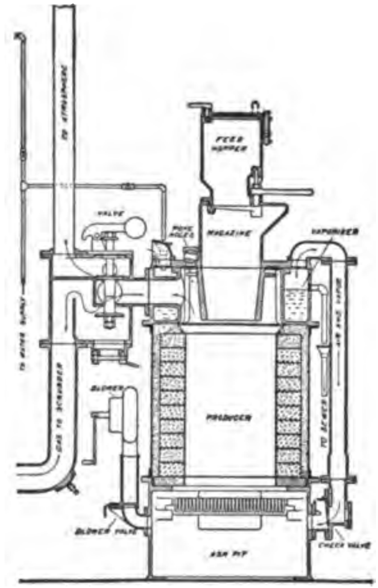


FIG. 7–13. — Fairbanks' Producer.

Riche. * Riche's producer, Fig. 7–14, is especially designed for wood, but it may also be used for hard coal and coke. The

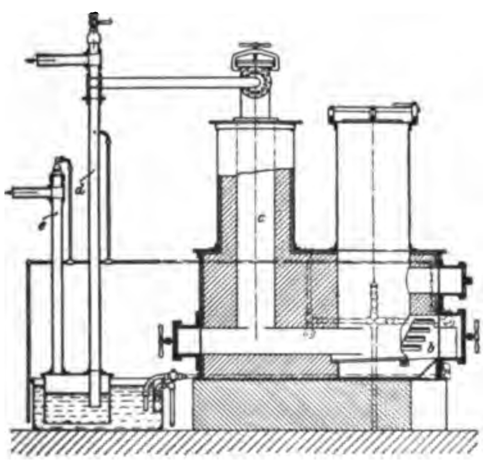


FIG. 7–14. — Riche's Producer.

* Génie Civil, 1901–2, p. 398.

column at the right serves as a magazine for the fuel bed at *b*. The gases formed are sucked up through *C*, which is kept filled with incandescent charcoal. The fixed gas passes through pipes *d* and *e*, being washed by means of sprayed water in its passage.

*Lencauchez.** Instead of using two separate stacks, the two fuels may be put together in one furnace, as is done by Lencauchez, Fig. 7-15. Long-flaming bituminous coal is charged through *a*, while through *b* coke from coke ovens or gas works is charged. Water is evaporated in the ash pit. The openings *c* and *d* show only black smoke, while *e* shows a colorless gas. The gases from the soft coal are fixed in passing through the incandescent coke layer. Schöttler mentions that this gas is not used as power

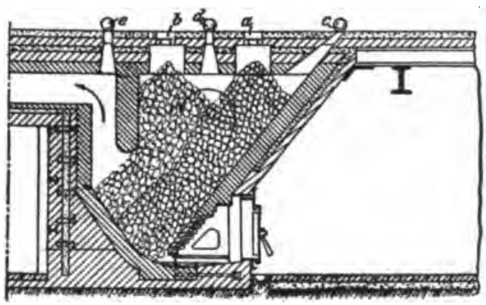


FIG. 7-15. — Lencauchez Producer.

gas but for heating purposes. The proportions of the charge are about 20–25 per cent coke to 80–75 per cent soft coal.

Lencauchez. † Another way of reaching the same end, that is, fixing the tarry gases, is by the use of so-called double generators. In these generators the charge burns both top and bottom, the gases being drawn off at about the middle of the producer. Fig. 7-16 shows a generator of this type designed by Lencauchez. It consists essentially of three conical parts. The upper one serves as a magazine. In the middle one the fire burns on top, in the bottom one at the bottom. Air enters at *D* and *B*. Water is introduced at *A* and evaporated in the ash pit. The air and gas currents are shown by the arrows. The gases of distillation formed in the magazine are drawn downward through the

* Zeitschrift des Vereins deutscher Ingenieure, 1905, p. 1903.

† Zeitschrift des Vereins deutscher Ingenieure, 1905, p. 1905

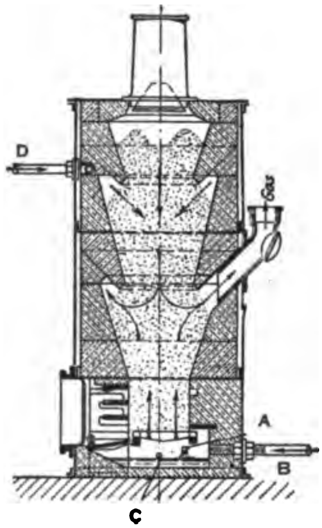


FIG. 7-16. — Lencauchez Double Zone Generator.

incandescent fuel in the middle section and are so fixed. Any unburned coked fuel then passes on toward the grate and is gasified on reaching the incandescent zone in the lower section.

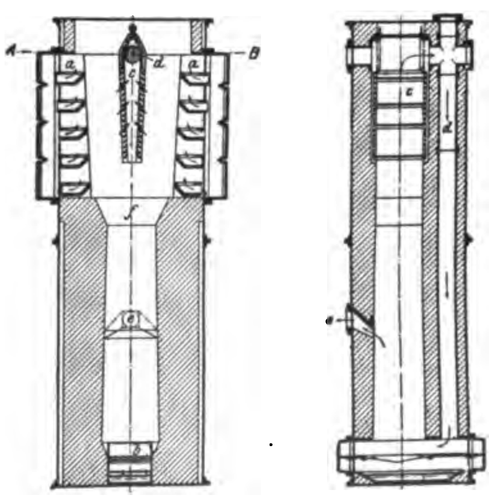


FIG. 7-17. — Koerting Producer for Peat.

*Koerting.** Koerting's producer for peat, Fig. 7–17, is built along lines similar to the above. The producer consists of two parts, of which the upper and larger one is fitted with step grates, *a a*, the lower one with an ordinary grate *b*. When in operation, the charge of peat in the upper parts burns only near the grates *a a*, while the inner core of peat is only coked. The gases so formed pass into the perforated pipe *c* and are led through *d* under the grate *b*. The coked peat formed in the upper part passes downward and is gasified over the grate *b*. The gases from *d* pass through the incandescent layer above *b*, are fixed, and together with the gases formed over *b* pass out at *e*. One would naturally assume at first sight that the gases of distillation would pass directly downward and out at *e*, instead of entering *c* and passing through *d*. The direction of the movement is regulated by creating a slight vacuum under the grate *b*, and by contracting the producer section at *f*. This contraction produces an extra resistance to the passage of gases downward, and they consequently take the more convenient way through *c* and *d*. Schöttler gives the analysis of a peat gas so made as

14 per cent CO_2, 4 per cent CH_4, 15 per cent CO, 10 per cent H, and 57 per cent N,

producer efficiency being about 75 per cent.

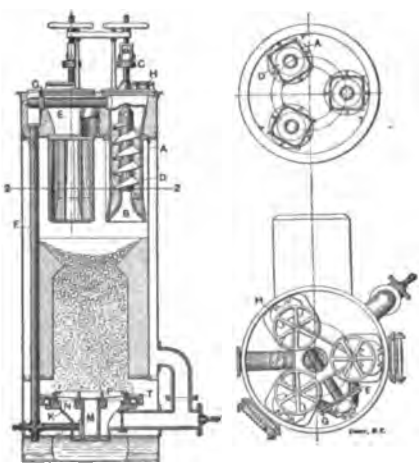


FIG. 7–18. — Crossley Producer.

* Zeitschrift des Vereins deutscher Ingenieure, 1905, p. 1909.

COMBINATION SYSTEMS. — *Crossley.* The Crossley producer, Fig. 7–18, is suitable for bituminous coal, or other fuel high in volatile matter. The fuel is first charged into the retorts A, and is there distilled by the sensible heat of the gases formed below. When charging, the valve B is drawn tightly against its seat, and the fuel is projected downward by turning the spiral D by means of the capstan. When the distillation is complete the casting B is lowered and the coked fuel is thrown into the main part of the producer by turning D. At the same time it is broken up by the spirals. The gases of distillation are drawn out of the retorts through the pipe F, and pass under the grate T, at M. The suction is produced by a fan blower, which at the same time draws air and steam to the main bed of fuel, and discharges the resulting producer gas into a holder at suitable pressure.

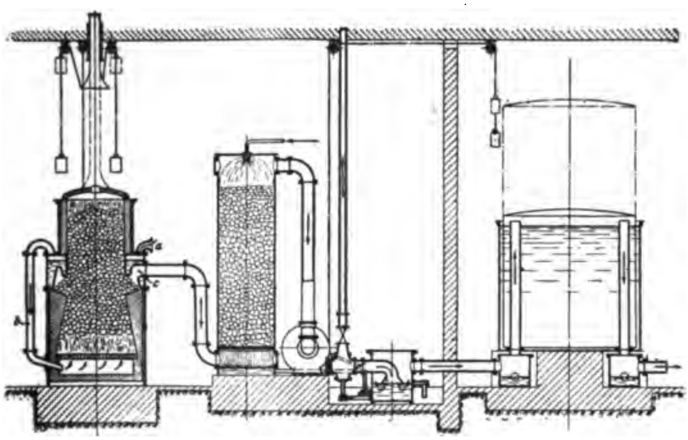


FIG. 7–19. — Deutz Double Zone Producer.

Deutz. The Deutz plant for bituminous coal is shown in Fig. 7–19. The producer is of the double-generator type. The air enters the vaporizer at a, and reaches the ash pit through pipe b. The fire burns top and bottom. Green coal is charged at the top. It is distilled, the gases formed pass downward through incandescent fuel and are so fixed, passing out at c. The greater part of the fuel coked near the top passes downward and is finally gasified in the lower part of the producer, the gases also passing out at c. Suction is maintained by a fan connected beyond the coke scrubber. Owing to the fixing of the tarry gases it is found

that a coke scrubber is all that is required. The fan, which is almost a necessary part of such a plant, owing to the increased friction entailed by the design of the producer, delivers the gas to a holder. The holder regulates the make of gas by controlling the fan. During the firing-up period, a purge pipe is lowered over the top of the generator, which has no special charging bell. The same fan is then used as a blower to furnish air to the producer, by-passing the scrubber.

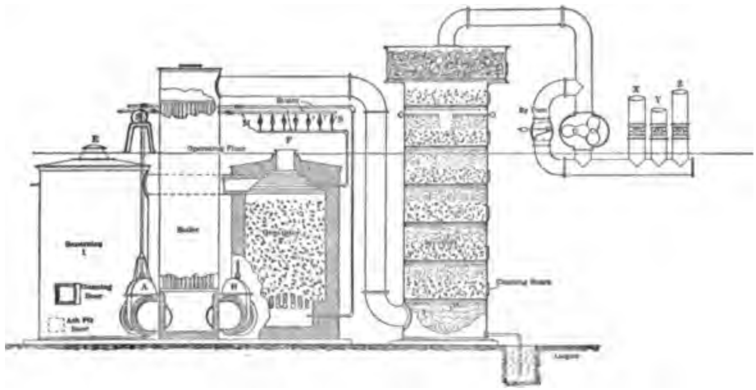


FIG. 7-20. — Loomis-Pettibone Producer.

*Loomis-Pettibone.** The Loomis-Pettibone gas apparatus, Fig. 7-20, may be used for the making of producer gas alone, or for alternate manufacture of water gas and lean power gas. Each plant consists of two producers, a boiler, a combination wet and dry scrubber, and an exhauster which delivers the gas made to the proper holder or the purge pipe, depending upon the positions of the valves x, y and z. The producers are charged through the top. Through the same doors the air also enters. Thus the gases pass downward through incandescent fuel, so that any kind of fuel may be employed. Leaving the producers at the bottom, the gases pass through the boiler, where they are cooled, and then through the scrubber and purifier. The make of gas is controlled by the speed of the exhauster, the suction pressure being ordinarily from 12 to 24 inches, the delivery pressure about 6 inches of water.

Suppose it is desired to make water gas. The feed doors E

* The Power Plant of the Montezuma Copper Co., John Langton.

and *F*, and the valve *B*, are closed. Steam is then admitted under the grate of producer No. 2, the right-hand producer in Fig. 7-20. The water gas formed in No. 2 is passed through the top connection to No. 1, passes down No. 1, out at *A*, and thus up the boiler and through the scrubber to gas holder. The next time water gas is run, the directions are reversed, or this may be done when one half of a water-gas run is completed. To make water gas, the aim is to blow the producers up with air as hot as possible, and then pass steam only through them until a great deal of it goes through undecompósed. Then air alone is again used to get the fires hot. This procedure means a wide range of temperature in the producer, and while this does not affect the quality of the water gas seriously, it does affect the producer gas made during the blowing-up periods, as a great deal of CO_2 is formed when the producers are comparatively cool. In a large plant where several producers are at one time delivering to the same gas holder, this irregularity is not much felt at the mixing valve of the engine where the two gases are mixed in proper proportion. For a single set plant this feature of varying producer gas make is more serious, and for this case the producers may be operated in a different manner.

To this end the steam connections, *M*, are used so that steam may be introduced along with the air, thus making water gas and lean gas concurrently, maintaining the producer temperature fairly constant for a long time; in other words, true producer gas is made. To get around the too fine adjustment of air and steam for constant temperature, somewhat less top steam than could be used is employed, thus allowing the temperature to slowly rise. A short water-gas run brings it down again to the proper point. The water gas made is slowly fed from its holder to the main gas holder through a controlled throttle valve so that its rate of feeding approximately equals its rate of production.

4. Some Producer Details. — Mathot, in *The Engineering Magazine*, May, 1905, gives the following data for suction porducers:

To get the greatest production of H and the most effective reduction of CO_2, make the cross-section of the base of the producer from 0.6 to 0.9 of the piston area of the engine. This is for a single cylinder 4-cycle engine at from 600–800 feet piston

speed. The depth of fuel bed should be from 3 to 5 times the diameter at the base, for ½-inch to ¾-inch lump coal. Amount of water dissociated is from 0.8 to 1.2 times the weight of coal consumed.

More than twenty analyses give the following average figures:

CO, 24 per cent; CO_2, 5 per cent; H, 17 per cent; N, 54 per cent.

Calorific power from 135 to 150 B. T. U. per cubic foot. Average coai used shows 89 per cent C, 2 per cent H, 4 per cent O + N, and 5 per cent ash. Best size coal is from ½-inch to 1¼-inch lump with 8 to 10 per cent of ash, and not more than 8 to 10 per cent volatile matter.

Water in the ash pan is satisfactory for steam production if the air is pre-heated. But preference is now given to internally heated vaporizers. Tubular vaporizers produce sufficient steam after 10 to 13 minutes. With well-designed apparatus, any suction plant should be in operation 25 minutes after lighting up.

The volume of the scrubber should be from 6 to 8 times that of the producer. Its height should be from 3 to 4 times its diameter. The filling is coke in pieces 3 to 4 inches in diameter, the coarser pieces near the bottom, the smaller near the top. Amount of water used for washing is from 3 to 5 gallons per horse-power per hour for anthracite gas.

According to the DeLavergne Company, the impurities to be removed from one ton of fuel are, for anthracite, from 1 to 2 pounds of ammonia, traces of sulfur, and from 5 to 10 pounds of tar; for bituminous coal, from 4 to 5 pounds of ammonia, sulfur from traces to 5 per cent, and from 10 to 12 gallons of tar.

The Morgan Construction Company point out the effect of sulfur on the formation of clinker. A coal with a large per cent of ash may work satisfactorily in a producer provided the sulfur does not exceed 1 per cent. Above 3 per cent the effect of sulfur in forming clinker is badly felt unless special facilities are at hand to break up such formations.

The same company makes the following statements regarding pressure-producer capacities. The best rate of combustion seems to be about 10 pounds of bituminous coal per square foot cross-section per hour, the coal carrying 10 per cent of ash and 1½ per cent of sulfur. With coals of lower ash content the rate

may be increased to 12 pounds, and in some special cases to 15 pounds. For poorer coals, however, the rate may sink to 6 or 8 pounds. Regarding the size of gas flues and pipes, the statement is made that each square foot of gas flue will take care of a gasification of 200 pounds of coal per hour, and will serve a gas-making area of 16 square feet in the producer.

CHAPTER VIII

THE GAS-ENGINE FUELS: LIQUID FUELS, CARBURETER AND VAPORIZERS

1. **The Crude Oils and their Distillates.** — The crude petroleum oils are practically mechanical mixtures of various hydrocarbon families. It is therefore to be expected that crude oils from various fields show compositions considerably different. From these crude oils we obtain by distillation all of the mineral oils now so widely used for a great variety of purposes. Of these oils the various gasolines, kerosenes, and the so-called "distillates," besides crude oil itself, are used for gas-engine fuels.

CRUDE OIL. — The following table* gives a few analyses of crude oils from different parts of the world. These oils are usually of a dark green color, the specific gravity is between 0.80 to 0.90 at 32 degrees, and the flash point between 76 and 93 degrees Fahrenheit.

	Sp. Gravity at 32°	C	H	O and Impurities	Lower Heating Value per pound
Heavy crude W. Virginia	.873	83.5	13.3	3.2	18324
Light crude W. Virginia	.841	84.3	14.1	1.6	18400
Heavy crude Pa.	.886	84.9	13.7	1.4	19210
Light crude Pa.	.826	82.0	14.8	3.2	17930
Heavy crude Ohio	.886	84.9	13.7	1.4	19210
Salt Creek, Wyoming	——	——	——	——	19463
Bothwell, Canada	.857	84.3	13.4	2.3	20410
Lima, Ohio	——	80.2	17.1	2.7	21600
Schwabwiller, Lower Rhine	.861	86.2	13.3	.5	18781
Gallicia, East	.872	82.2	12.1	5.7	18010
Gallicia, East	.885	85.3	12.6	2.1	18416
Light Crude, Baku	.884	86.3	13.6	.1	19440
Heavy Crude, Baku	.956	86.6	12.3	.1.1	19496
Keudong, Java	——	85.0	11.2	2.8	18330

* Poole, the Calorific Power of Fuels.

GASOLINE. — After the distillation of the very light products from crude oil, we obtain a series of gasolines of varying flash points and gravity. The first of these is 86 degree gasoline, as measured on the Beaumé scale. This forms a mixture with air so readily even at ordinary temperature that it is somewhat dangerous and not often used. The next gasoline, the original liquid gas-engine fuel, is 74-degree gasoline. Owing to the much greater consumption of gasoline, due to the introduction of the automobile mainly, the gravity of gasoline and also the flash point have slowly gone up, until to-day 69-degree gasoline is quite common.

Gasoline vaporizes readily at ordinary room temperatures, and it is therefore necessary to keep it covered, not only to prevent loss but also accidents due to explosions. Insurance companies usually specify that any quantity of it must be kept in an underground tank outside of the building, and that is undoubtedly the best way.

Data on the heating value of gasoline is not at all plentiful. A sample the writer had the opportunity of testing recently gave the following figures:

Specific gravity, Beaumé 69.5 at 60 degrees Fahrenheit.

Composition, 84.76 per cent C, 15.24 per cent H.

Lower heating value as computed from analysis, 20411 B. T. U.

Heating value as determined by Junker's calorimeter, higher value 19606, lower value 18482 B. T. U.

It should be noted that this is one of the instances where the heating value as computed is much higher than the actual value, nearly 2000 B. T. U. = 10 per cent in this case. It shows the necessity of calorimetric determination for accurate work.

KEROSENE. — Kerosene, the next heavier distillate beyond gasolines, is not as extensively used as gasoline in this country for gas engines. It will not form an explosive mixture with air at ordinary temperatures, and therefore requires more elaborate apparatus for the formation of such a mixture. The following data is given by Mr. S. A. Moss for ordinary American kerosene:

Sp. Gr. at 60°	Flash point open	C by weight	H by weight	Lower Heating Value per lb. B. T. U.
.80	100°	.85	.15	18520

Further reliable data on kerosene is given in a lecture by Diesel in 1897, in connection with Schöttler's test of the Diesel engine.

Specific gravity at 24°C		(75.2° F.) .789.
Average composition,	C	85.13 per cent.
	H	14.21 per cent.
	O	.66 per cent.

Lower heating value computed from analysis, 19874 B. T. U.
Lower heating value by Junker's calorimeter,
average of five tests, 18242 B. T. U.

Distillates. — A distillation product resembling kerosene in its general properties is sometimes used as fuel. These so-called distillates are not as well refined as kerosene, but are handled the same when used for engine work.

The following table, due to Hofer and transposed from Güldner, shows a good method of presenting the various distillation products from crude oil:

(*a*) *Volatile Oils.*

Sp. Gr., .65–.75, Flash Point Below 70° F., Boiling-Point Below 300°

	Sp. Gr.	Boiling-Point
Petroleum Ether	.65–.66	95°–122° F.
Benzine	.66–.68	112°–158°
Various Gasolenes	.67–.74	149°–300°

(*b*) *Illuminating Oils.*

Sp. Gr., .78–.86, Flash Point 70°–158°, Boiling-Point above 300° F.

	Sp. Gr.
American Kerosene	.78 –.81
Russian Kerosene	.82 –.825
Standard White	.808–.812
Prime White	.80 –.805
Astralin	.85 –.86

(c) *Heavy Oils.*

SP. GR., .86–.96, FLASH POINT 374–482° F.

	Sp. Gr.
Solar Oil	.86–.88
Lubricating and Cylinder Oils.	.88–.90, etc., can be used as lubricating oils only.

2. **Alcohol.** — The use of alcohol as a gas-engine fuel in this country is as yet of no great importance, although the recent action of Congress in removing internal revenue under certain restrictions will do a great deal toward helping alcohol to the place it deserves as a fuel. In some European countries, Germany for instance, the price of alcohol is not much greater than that of gasoline, and it may therefore compete with gasoline with some success, especially when some of its advantages are considered. It is much safer than gasoline as regards fire risks, and since it always contains some water, a higher degree of compression may be employed in the engine, guaranteeing better thermal efficiency. These advantages compensate largely the greater specific heat cost of alcohol.

Ethyl alcohol,* whose chemical formula is C_2H_6O, may be made in various ways, but the commercial alcohol of to-day is the result of fermentation, generally of grape sugar, in the final stage. The raw materials are various. Thus, according to Sand † they may be divided into three classes:

1. Those containing starch — potatoes, with 15 to 24 per cent starch.
   rye, with 50 to 56 per cent starch,
   corn, with 60 per cent starch,
2. Those containing sugar — sugar beet, with 8 to 18 per cent sugar,
   sugar cane, with 12 to 16 per cent sugar.
3. Those containing alcohol—wine with 9 to 16 per cent alcohol.

* What follows is a reprint from an article by the writer on the use of alcohol as a fuel for gas engines in Marine Engineering, June, 1906.

† Sand, Zeitschrift des Vereines deutscher Ingenieure, 1894, p. 933.

The method of manufacture, of course, varies with the raw material, but need not be described in detail here. Theoretically, 100 pounds of grape sugar should yield 51 pounds of pure alcohol; in reality the yield is from $\frac{1}{5}$ to $\frac{1}{3}$ less than this amount.

A second method of producing alcohol, notably mentioned by Witz in his "Moteurs à Gaz et à Pétrole," is to start with calcium carbide as a raw material. This, by a somewhat complicated process, can be changed from CaC_2 through the stages of C_2H_2 and C_2H_4 to alcohol, C_2H_6O. Barium carbide or strontium carbide can be used in the same way. Witz states that from 1 kilogram (2.2 pounds) of calcium carbide 0.8 liter (1.69 pints) of alcohol can be obtained; this is equivalent to 0.096 United States gallons of alcohol from 1 pound of carbide. Estimating the price of carbide at 3 cents per pound, which is even now somewhat below the market price, 1 gallon of alcohol would therefore cost, in raw material alone, 31.2 cents, to say nothing of the cost of the chemical operations. The by-products in the case of the calcium carbide do not amount to much. There is consequently little likelihood that the so-called synthetic or mineral alcohol will ever seriously compete with gasoline or kerosene for power.

The heating value of alcohol cannot be accurately computed from its chemical composition, because nothing definite is known of the arrangement of the atoms entering the composition. We therefore have to depend upon the calorimeter. The figures determined for absolute alcohol by various experimenters are as follows:

	Higher Heating Value per pound	Lower Heating Value per pound
Thomson	13310 B. T. U.	12036 B. T. U.
Favre & Silberman	12913 B. T. U.	11664 B. T. U.

The value 11664 B. T. U. is the one most generally used. Absolute alcohol has a specific gravity of 0.7946 at 15 degrees Centigrade (59 degrees Fahrenheit), so that one gallon of pure alcohol weighs 6.625 pounds, and has a lower heating value of 77274 B. T. U.

One pound of C_2H_6O contains 0.522 pound carbon, 0.130 pound hydrogen, and 0.348 pound oxygen.

According to this there will be required for the combustion of one pound of absolute or 100 per cent alcohol,

$$\frac{(0.522 \times 2.66) + (0.130 \times 8) - 0.348}{0.23} = 9 \text{ pounds of air.}$$

This is the equivalent of 111.5 cubic feet of air at 62 degrees Fahrenheit, per pound of C_2H_6O. Commercial alcohol, however, is never pure, but nearly always contains a certain quantity of water, the admixture being measured according to volume per cent. Thus, 90 per cent alcohol means that the mixture carries 10 per cent by volume of water. The heating value of such alcohol is of course correspondingly reduced from that of 100 per cent alcohol according to the following table, due to Schöttler:

Absolute alcohol volume—per cent	Specific gravity	Absolute alcohol weight—per cent	Lower heating value per pound B. T. U.
95	0.805	93.8	10880
90	0.815	87.7	10080
85	0.826	81.8	9360
80	0.836	76.1	8630
75	0.846	70.5	7920
70	0.856	65.0	7200
70	0.856	65.0	7200

It is required by law, in countries where alcohol is now used in the industries, to so fix the fuel that it is rendered undrinkable. This process is called "denaturizing" the alcohol. The bill passed by Congress at the last session provides for the same thing. The materials used for this purpose differ in the various European countries. Some of them try to keep the process a secret, hence some of the information given in the following table * is based upon analyses.

* Zeitschrift des Vereines deutscher Ingenieure, June, 1905.

MATERIALS USED TO DENATURIZE ETHYL ALCOHOL

Country	Sp. Gr. of Denat-urized Alcohol at 15° C	Methylene (wood alcohol) and its impurities per cent	Pyridine or Pyridine Bases per cent	Ace-tone per cent	Benzol per cent	Benzine per cent
France	0.832	7.5	—	2.5	—	0.5
Germany						
Denat. Alcohol	0.819	1.5	0.5	0.5	—	—
Motor Alcohol	0.825	0.75	0.25	0.25	2.0	—
Austria						
Denat. Alcohol	0.835	3.75	0.5	1.25	—	—
Motor Alcohol	0.826	0.5	trace	trace	2.5	—
Russia	0.836	10.0	0.5	5.0	—	—
Italy						
Motor Alcohol	0.835	6.5	0.65	2.0	1.0	—
Switzerland	0.837	5.0	0.32	2.2	—	—

It will be noted from the table that the material most used for denaturizing ethyl alcohol is wood alcohol. The heating value of the fuel is by the addition of the denaturizing liquid changed but little in most cases.

Benzol, C_6H_6, besides being used for denaturizing, is sometimes used in larger quantities than indicated in the above table, for the purpose of increasing the heating value of the fuel mixture per pound. Benzol has a specific gravity of 0.866 and a heating value of 17190 B. T. U. per pound. A mixture of x per cent by weight of absolute alcohol with y per cent of benzol, will therefore have a heating value of

$$[11664\,x + 17190\,y] \text{ B. T. U. per lb.}$$

If the alcohol is not absolute, its proper heating value should be substituted from the table above given. In this way from 10 to 40 per cent of benzol is sometimes employed, thus raising the heating value of the fuel, and at the same time decreasing the specific heat cost, *i.e.*, the cost per heat unit.

There is a second reason why benzol is employed. Under certain circumstances there will be formed acetic acid in the products of combustion of alcohol. This causes rusting of the engine parts. On examination it will be found that this is due to a combustion

with insufficient air supply, and the surest way to prevent rusting, therefore, is to use a good excess of air, and to have a perfect mixture. Under such conditions there will be no danger of corrosion. It is also found that a good addition of benzol acts as an additional safeguard. It should not be forgotten in this connection, however, that the great advantage possessed by alcohol in its odorless exhaust is sacrificed to some extent by the use of benzol.

3. **Mixing Devices for Liquid Fuels.** — As has been already pointed out, the fuel mixture of a gas engine is always a gas or vapor mixed with air. Hence in the case of liquid fuels special devices are required to convert these fuels into gases or vapor. Such devices are indiscriminately known as carbureters, vaporizers, mixers, or mixing valves. In the strict sense, a carbureter is a device in which the mixture is formed by passing air over or through the liquid fuel. When no special heating of the fuel is done, these devices are applicable to the more volatile fuels only, as gasoline. The name carbureter, however, is also used when in this device gasoline is mechanically atomized or sprayed into the current of incoming air. The term vaporizer is usually employed when considerable heat for gasification or vaporization is required, as is the case for kerosene, crude oil, and alcohol.

MIXING DEVICES FOR GASOLINE: CARBURETERS. — Carbureters in which the air is passed over or through gasoline have the drawback that when the carbureter chamber is filled with fresh gasoline the lighter constituents of the liquid distil off first. The parts harder to vaporize remain behind, so that the fuel mixture is anything but constant, growing leaner as the vaporization progresses.

Atomizing or spraying carbureters are not open to this objection, a definite amount of gasoline being injected each time. Hence they are much used, especially in automobile work.

Figure 8-1 shows a carbureter which is of what has been very aptly termed the bubbling type. The suction stroke of the engine establishes a partial vacuum in *a* through the check valve *e*, and the chamber *d*, interposed for reasons of safety. This causes air to enter through the screen *b*, rising through *c* in a spray. Bubbling up through the gasoline it carries with it some of the vaporized liquid and goes through *d* and *e* to the engine,

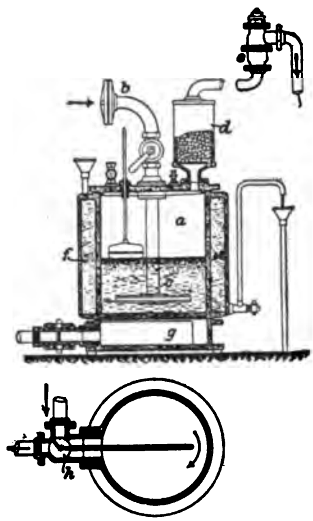


Fig. 8-1.

where a normal combustion mixture is formed by the addition of more air. The jacket water from the engine may be circulated through *f*, assisting in the vaporization. In special cases a part of the exhaust gases may also be circulated through the bottom, *g*.

Figures 8-2 and 8-3 are properly termed surface carbureters. In the former, the Reithman carbureter, the gasoline is fed from the reservoir *a* in drops into the chamber *b*, where it is absorbed by broad suspended wicks. From the surfaces of these the gasoline vaporizes, adding itself to the air which strikes through chamber *b* on every suction stroke. Chamber *f* is filled with clean gravel or wire

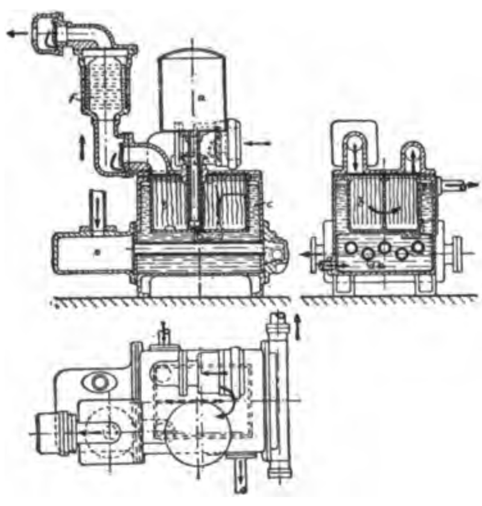


Fig. 8-2 — Reithman Carbureter.

screens, to act as a safety device against possible back firing. Chamber *b* is surrounded by a water jacket, the water in which is heated by the exhaust gases passing through the tubes *d d*.

The Petreano carbureter, Fig. 8–3, will serve for either gasoline or alcohol. The exhaust gases pass through a central pipe, *r*, which is surrounded by a jacket, *V*. The pipe *r* has a covering, *d*, of spongy asbestos film which is constantly kept moist with the liquid to be vaporized. The liquid fuel enters by one opening in the top, the air by another. The chamber *V* has four cones, as shown, two of which are also partially covered with asbestos. By means of these cones the fuel vapor and air are thoroughly mixed before entering the chamber *M* and finally the suction pipe to the engine. The small openings, *o*, are for the purpose of drawing off the heavier parts of the liquid, those that vaporize less easily, if there be any, in order not to interfere with the regularity of carbureting.

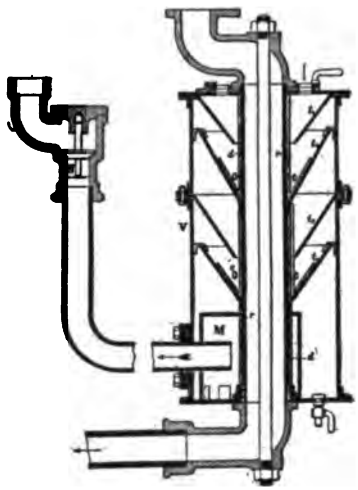


FIG. 8–3. — Petreano Carbureter.

Atomizing or spraying carbureters are, however, much more frequently employed, and the following to be described are all of this type.

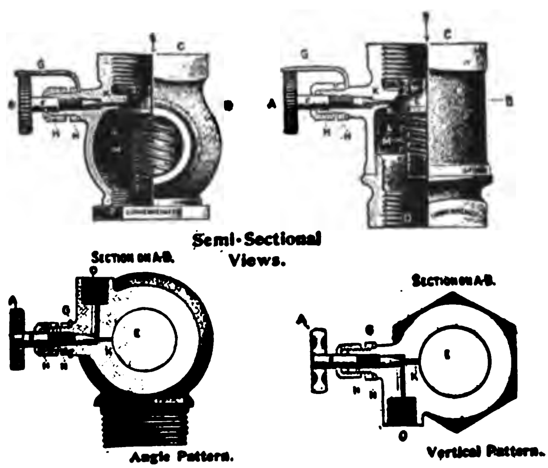


FIG. 8–4. — Lunkenheimer Mixing Valves.

The simplest design of this type of carbureter is shown in Fig. 8–4, which shows two of the Lunkenheimer carbureters. In either design, gasoline is furnished through the opening O, and its admission to the valve seat of the main suction valve E is regulated by the needle valve A. On the suction stroke of the engine the valve E automatically lifts, and the air rushing through this opening carries with it a certain amount of gasoline flowing from the small hole K. It is quite evident that this type of carbureter gives best service on hit-and-miss engines. For any other type a varying load would probably cause trouble, as there is no automatic regulation of the gasoline supply.

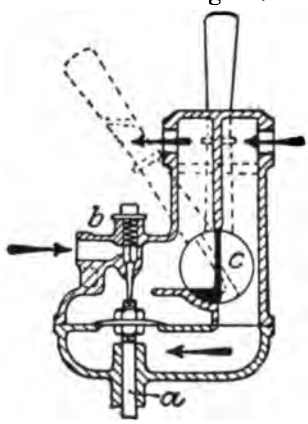


FIG. 8–5.—Sintz Carbureter.

Still of very simple design, but capable of regulation of the gasoline supply, is the Sintz carbureter, Fig. 8–5. The lift of the gasoline needle valve b is regulated by the lift of the mechanically regulated suction valve a. By means of the cock c it is possible to admit part of the air uncarbureted to the cylinder.

Figures 8–6 and 8–7 are examples of the so-called float-feed type of carbureters.

The Daimler, Fig. 8-6, is well known. A float, B, in the chamber, A, operates a pair of counterweight levers, E, and through them the valve spindle, D, which controls the admission of gasoline at C, keeping it at a constant level in the nozzle. Gasoline enters at N from a tank under slight pressure or at a higher level. O is a cloth filter and the plug P serves to catch any grit that may be brought in. The float chamber is vented through a small hole in the cap over the valve spindle to relieve any pressure which may be formed due to varying positions of the float B. The action of the carbureter is as follows: At each charging stroke of the engine air is drawn into the annular chamber H and passes with great velocity through the drop tube F, surrounding the gasoline nozzle. The gasoline is drawn in a jet from the nozzle and, with the air, striking the deflector K, the two are very thoroughly mixed, passing to the engine through M. An aux-

iliary air supply can be admitted through the cap at the top, the openings through which can be regulated.

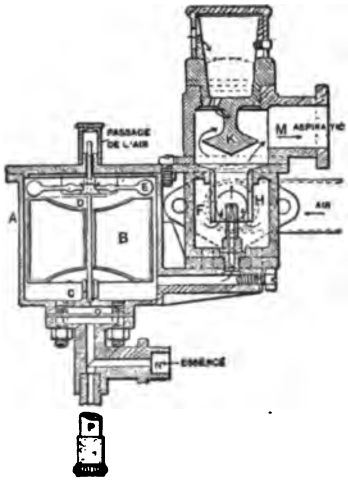


FIG. 8-6. — Daimler Carbureter.

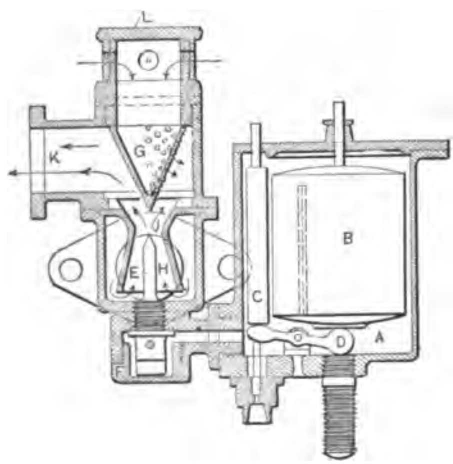


FIG. 8-7. — Abeille Carbureter.

The Abeille carbureter, Fig. 8–7, embodies the same idea as the Daimler. The float *B*, by operating the lever *D*, opens and closes a needle valve at the lower end of the weighted spindle *C*, to maintain a constant level just below the opening in the nozzle

E. On the suction stroke, air enters at *H*, and rushing up through the double cone draws the gasoline from the nozzle, atomizing it by striking the perforated cone *G*. A secondary air supply is admitted through the cap *L* and the holes in *G*, being regulated by the position of the cap *L*.

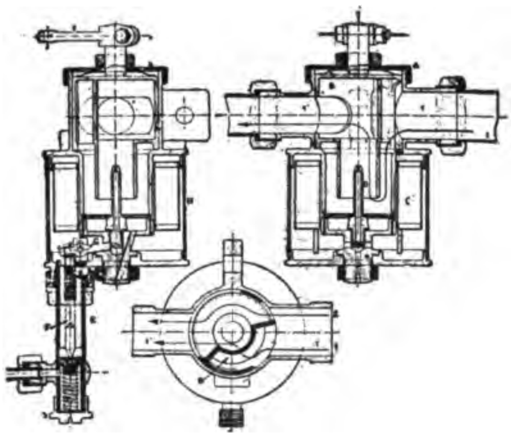


Fig. 8-8.—De Dion Carbureter.

Of more complicated design, but in principle the same as the other float-feed carbureters, is the De Dion, Fig. 8-8. Air is admitted through the tube *t*. By the position of the valve *V* (see horizontal section), which position can be regulated by the lever *I* (see vertical section), part of the air goes directly into the discharge pipe *t′*, while the rest is deflected downward to be carbureted. The course of this latter body of air is indicated in the right vertical section by the arrow. The level of the gasoline in the nozzle *D* is again kept constant by the annular float *C* in the chamber *H*, operating the lever *G* and through it the valve *F*.

It will be noted that none of the atomizing or spraying carbureters for gasoline so far described are heated in any manner. Occasionally, however, we find one in which the atomizing is assisted by heat. The W. Hay vaporizer, Fig. 8-9, is of this type. Gasoline enters the annular chamber, *a a*, through the pipe *d*. From this chamber a number of small openings lead through the seat of the suction valve *E*. Some of these openings are provided with adjusting screws as shown at the left. On

the opening of E, the inrush of air atomizes the gasoline flowing from these small openings, and the current of air and gasoline striking the wings e of the fan h, supported on the spindle j, sets the fan in motion, thus promoting a thorough mixture of air and vapor before the fuel finally passes through x to the admission valve A. The exhaust gases from B are passed through the chamber F, finally escaping through the slotted openings g. In their passage they heat both the gasoline storage chamber a a and the fan chamber.

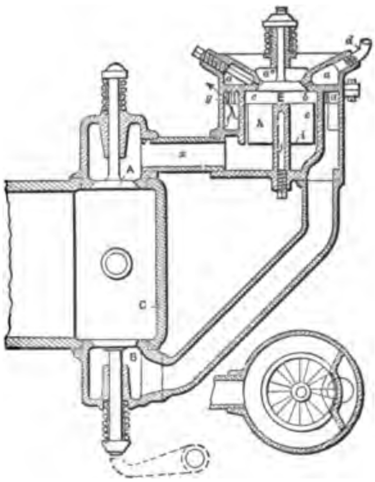


Fig. 8–9. — W. Hay Vaporizer.

Occasionally we also find carbureters which combine the several principles of those described. Thus in the Gautier carbureter, Fig. 8–10, the gasoline is admitted through A, the supply being regulated by the valve

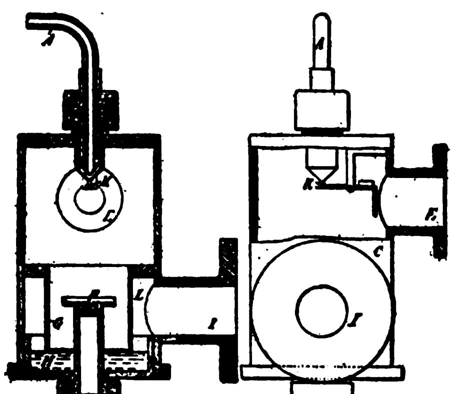


Fig. 8–10. — Gautier Carbureter.

K, which opens at the proper moment owing to suction through pipe E. Part of the supply falls on to the saucer F, and from there into the reservoir H. Into the liquid at H dips a pipe G,

supported as shown. The air supply through *I* circulates through the chamber *L*, bubbles through the liquid at *H* into the chamber surrounding the saucer *F*, licks up some of the gasoline in the saucer, and the mixture finally escapes through *E*.

VAPORIZING DEVICES FOR CRUDE OIL AND KEROSENE. — While it is fairly easy to satisfactorily atomize or vaporize gasoline and to maintain the mixture, the thing is a little more difficult with kerosene, and much more so with crude oil. In either case the agency of heat is required, and this is applied either in a separate vaporizer or retort, or the kerosene or crude oil is injected directly into the cylinder, the vaporization taking place in the combustion chamber.

One trouble with kerosene is the readiness with which some of the vapor is condensed on striking comparatively cool surfaces. This may happen on the mixture striking parts of the water-cooled walls. In such a case part of the fuel may go through the cylinder unburned, and this is a point that should be carefully guarded against when a special vaporizer for kerosene is employed.

In the case of crude oil, the heating in the vaporizer results in the distillation of the lighter products first. The amount of vapor formed will naturally be less and less as the distillation proceeds, resulting in a constant impoverishing of the fuel mixture. The remedy would therefore seem to be a constant supply of fresh oil and a removal of the old before it has commenced to seriously decrease its yield. This is what is actually done in some devices. The method, however, naturally results in the fact that the plant can only use part of the crude oil. It is claimed by some that only 10 per cent of the oil is so withdrawn from the vaporizer, but this seems extremely doubtful.

A crude oil-air mixture is open to the same objection as a kerosene mixture as regards condensation of some of the heavier hydrocarbons and consequent loss. Both of these mixtures are also subject to cracking; that is, a breaking up of the heavier hydrocarbons into the lighter with a consequent deposit of carbon.

For large power plants the best solution in the case of crude oil would seem to be the use of some type of oil-gas producer. Some of these are in actual use along the Pacific coast, and will be described later.

Figure 8–11 shows the kerosene atomizer used on the Hornsby engine. With it is combined the regulating valve. Kerosene is pumped up through the lower right-hand pipe in the sectional cut. The governor regulates through *c* the position of the overflow valve *b*, the surplus kerosene flowing back through *d* to the reservoir. The kerosene at the moment demanded by the engine

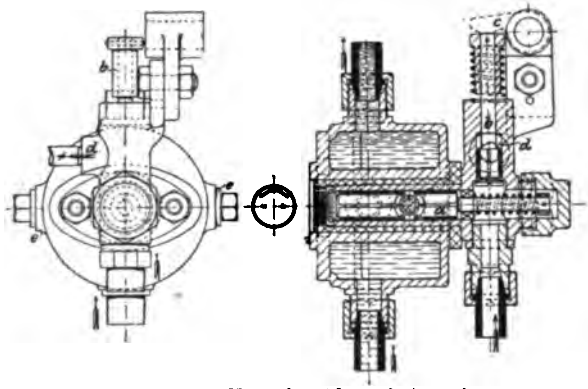


FIG. 8–11. — Hornsby-Akroyd Atomizer.

is forced by the pump through the plug *a*, issuing from its end in a fine spray. This plug is kept cool by a water jacket as shown.

Another atomizing kerosene carbureter is the Gibbon, shown in Fig. 8–12. In this a valveless pump *x w*, actuated by a cam, injects kerosene from the tank up through the atomizing opening into the chamber *U*. This is furnished with wings, *U′*, to present a larger surface for heating. Chamber *U* is directly connected with the combustion chamber, and is surrounded by a light case to prevent radiation. At the start it is heated by a lamp, but after a short period of operation the charge ignites by compression, as is the case also in the Hornsby engines.

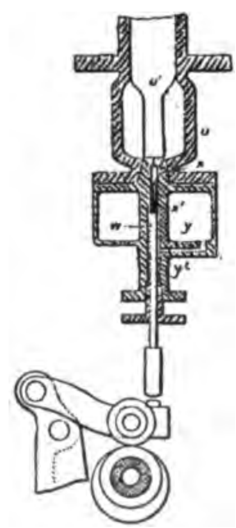


FIG. 8–12. — Gibbon Kerosene Vaporizer.

In the Crossley vaporizer, Fig. 8–13, a certain fixed amount of oil is measured and drawn into the vaporizer by the air on the opening of the valve. In this case ignition

is produced by means of a hot tube, and the engine is of course governed on the hit-and-miss system. The lamp heating the hot tube at the same time heats the vaporizer.

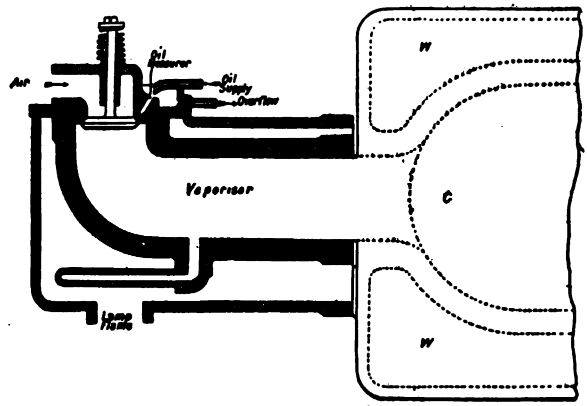


FIG. 8-13. — Crossley Vaporizer.

The Priestman engine is designed for either crude oil or kerosene. Its vaporizer is shown in Figs. 8–14 and 8–15. Oil is admitted to the spraying nozzle K from a reservoir under pressure.

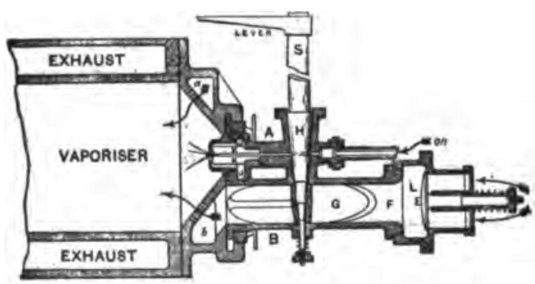


FIG. 8–14. — Priestman Vaporizer.

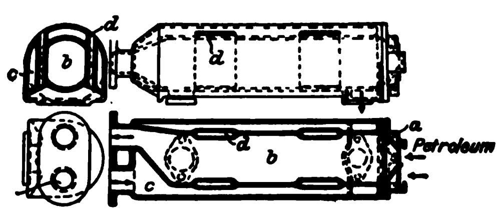


FIG. 8–15. — Priestman Vaporizer.

The pump furnishing the air for this purpose also furnishes air to the annular space *J*, surrounding the nozzle *K*, for the purpose of atomizing the oil. The finely divided oil enters the vaporizer chamber and, mixing with the larger body of air admitted on the suction stroke to the vaporizer chamber through the valve *L*, passes on to the cylinder. The vaporizer chamber, Fig. 8–15, is heated by the exhaust gases passing through the jacket space *C*, thus vaporizing the oil spray in its passage. On starting, the vaporizer chamber is heated by a lamp whose hot gases pass through the jacket spaces *d*. The governor, by regulating the position of the plug valve *H*, regulates at the same time both the oil and the air supply to the vaporizer.

A crude oil vaporizer quite extensively used on the Pacific coast is the "Economist" retort, Fig. 8–16. This consists of

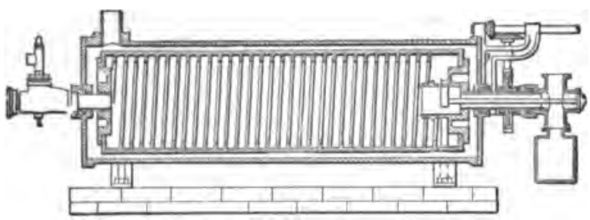


Fig. 8–16. — "Economist" Crude Oil Vaporizer.

two concentric shells as shown. In the jacket space between them the engine exhaust gases are circulated to furnish the heat for vaporization. The inner drum is slowly rotated upon its axis, as shown by the gearing at the right, and it is furnished on the interior with a continuous spiral rib. The crude oil is admitted through the central pipe at the left. By the rotation of the drum it is carried up the sides, turned over and over, thus exposing a thin sheet all the time to the action of the heat, which is most favorable for complete vaporization. At the same time, owing to the action of the spiral rib, the oil is slowly carried forward through the drum. The residue is finally discharged through an outlet at the right, in a manner not clearly shown. The makers claim that during the progress of the oil through the drum the temperature is maintained just high enough to get all the gas from the oil. The method of regulation by this device is not clearly stated. If it is done by regulating the oil

supply to the retort according to the load, there would be considerable lag in the regulation; if done by regulating the air supply through the retort there would be a large variation in the composition of the mixture. An auxiliary air supply between generator and engine would help, but whether this is used or not is not stated.

Retorts as above described are made up to from 125 to 150 horse-power. Beyond this they become unsatisfactory, and other devices have to be employed. The one that seems to promise best at present is an oil-gas generator of the type of the Lowe.

The principle of the Lowe system is to heat up to a very high temperature a mass of firebrick checker work contained in a firebrick-lined steel shell, by means of a crude oil-air blast. When the desired temperature is reached the chimney connection to the generator is closed off, and crude oil and superheated steam in an intimate mixture are sent down through the highly heated checker work. The result is the formation of an impure oil water gas with a good deal of lampblack as a by-product. This by-product may be utilized as fuel to furnish the power necessary for blowing engines, etc., around the plant. The gas itself is washed in the ordinary manner, and is a high-grade gas of good illuminating power. The production of this gas is therefore carried on in the two stages, a heating-up period and a period of make. From published figures the efficiency of these oil-gas producers is as yet not very high. In the *Journal of Electricity, Power and Gas*, September, 1904, a consumption of 11.22 gallons of oil per 1000 cubic feet of gas made is given as the average for one plant for July. Other plants have shown 9 and 10 gallons per 1000 cubic feet of gas of about the same heating value. Assuming the crude oil to weigh 7 pounds per gallon, its heating value at 19000 B. T. U. per pound, and the heating value of the gas at 650 B. T. U. per cubic foot, the efficiency of generation on cold gas would be for the most favorable case above given

$$\frac{1000 \times 650}{9 \times 7 \times 19000} = 54.3\ \%$$

Vaporizing Devices for Alcohol. — The alcohol engine has perhaps received its greatest development in Germany.

It is for that reason that we shall have to turn to the literature of that country for the best and latest information on the details of alcohol engines.

The following material is a reprint from an article published by the writer on the alcohol question in *Marine Engineering*, June, 1906. The information is mainly due to the work of E. Meyer and of A. Schöttler, also to the discussions by Güldner, Diesel, and others. *

Regarding the formation of the fuel mixture with alcohol, it is found that it is less volatile than gasoline, but easier to handle than kerosene. In nearly all of the vaporizing devices for alcohol now on the market, the agency of heat, usually the exhaust heat of the waste gases, is used to aid in the formation of the mixture. This scheme has the drawback that no heat is available at the start when the engine is cold. To avoid an open flame for the purpose of heating the vaporizer at the start, which is both dangerous and cumbersome, the engines in most cases are started with gasoline, and when, after a few strokes, enough heat is available, the change is usually made by throwing over a single lever. In tests of ten different engines made by Meyer, it was shown that this change to alcohol could be made in the slowest case in 6 minutes and 40 seconds, the time of the fastest being 55 seconds.

Based on the manner of heating the vaporizer, we can distinguish the following classes:

1. Those in which no heat is employed.
2. Those in which the air is pre-heated.
3. Those in which the mixture is heated and superheated.

Of the first type is the Deutz, Figs. 8–17 and 8–18. When the engine is regulated by the throttling method, and not by the hit-and-miss system, it has been found that no pre-heating of air or fuel mixture is required. The reason for this is undoubtedly that in a hit-and-miss engine, under less than normal load, a succession of misses cools the cylinder down so far as to throw down some of the alcohol vapor on the next explosion, unless it is superheated. The Deutz engine is governed by throttling. The inlet valve *f* is actuated, through the levers

* E. Meyer, Zeitschrift d. V. d. I., 1903, pages, 513, 600, 632, 669; R. Schöttler, Zeitschrift d. V. d. I., 1902, pages, 1157, 1223; H. Güldner, Zeitschrift d. V. d. I., 1902, page 623; Diesel, Zeitschrift d. V. d. I., 1903, page 1366.

shown, by the cam *a*, which is of taper form and under the control of the governor Fig. 8–17. Upon the position of *a* depends the length of time the valve *f* is open. Through the bell crank *c d e* the cam also acts upon the plunger of the fuel pump *h*, operating in such a way as to cause suction during the first part of the cam movement, and pumping of the liquid during the second. Thus the fuel is injected during the second half of the suction stroke only, insuring a rich mixture around the igniter.

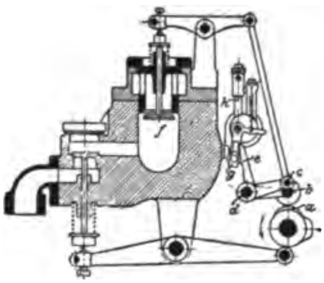


Fig. 8-17. — Deutz Alcohol Vaporizer.

The alcohol is forced through the sprayer or atomizer *i*, Fig. 8–18, into the current of air which enters through the valve *k*. Thus no pre-heating whatever is done, but the atomizing is thorough; and the ports into the cylinder are as direct and short as possible, hence no vapor is thrown down.

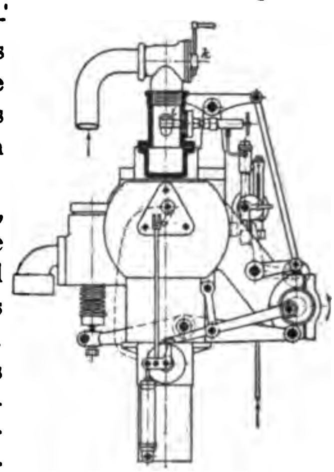


Fig. 8-18. — Deutz Alcohol Vaporizer.

The Altman vaporizer, Fig. 8–19, is of the second class. The air pipe *a—b* is surrounded at its lower end by the exhaust pipe; the air is thus pre-heated by the exhaust gases. A regulating valve for the air is placed at *e*. This, when drawn upward, decreases the amount of air passing, but always makes the air current strike through the upper part of the pipe, in this manner directing it always against the fuel nozzle *d*. The inlet valve *c* is operated by the lever *f*, actuated by the cam *l*, through the pendulum hit-and-miss governor *o m p*. This valve lever *f* at the same time opens the fuel valve *d*, through the reach rod shown and the finger *h i*, Fig. 8–21. How this is done is shown in Fig. 8–20. The lever *f*, on being depressed, forces down the point of the screw *k*, thereby turning the reach rod about its axis, which depresses

the point i, Fig. 8–21, opening the valve d. The amount of opening depends upon the position of the screw k, and this can be very finely adjusted by the worm and wheel arrangement shown. In this vaporizer the fuel supply is atomized partly by the current of air, and is afterwards vaporized by the heat of the pre-heated air.

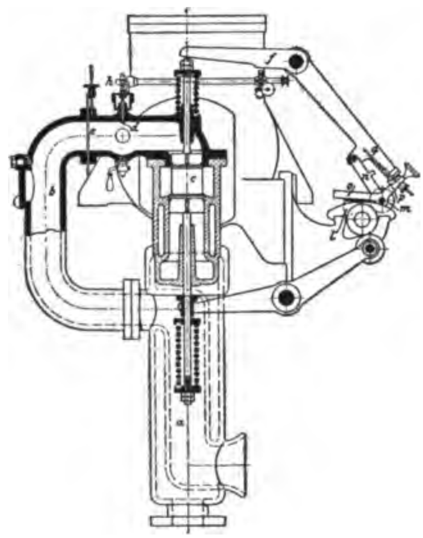


Fig. 8–19.—Altman Alcohol Vaporizer.

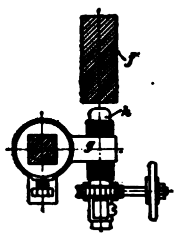


Fig. 8–20.

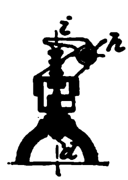


Fig. 8–21.

The following three vaporizers are of the third class. Fig. 8–22 shows the Swiderski-Longuemarre. Here also the exhaust gases are used for heating. They pass through the annular chamber a, and their action is aided by the radiating webs b b. The float d maintains a constant level in the supply chamber. From this chamber the flow of alcohol is regulated by the needle valve f.

The liquid flows into the space g, and overflows through a number of small openings h h. Air entering through i is made to pass partly outside, partly inside, the concentric spaces created

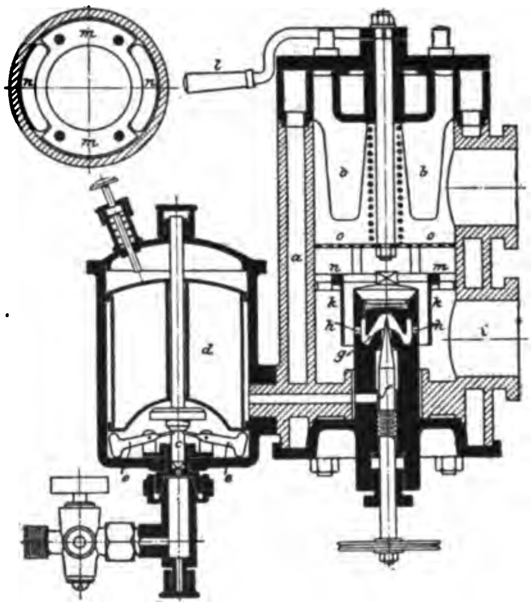


FIG. 8-22. — Swiderski-Longuemarre Alcohol Carbureter.

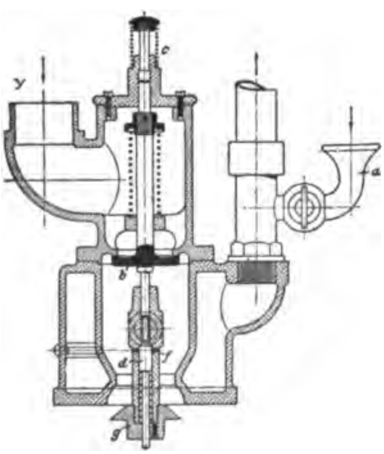


FIG. 8-23. — Dresden Alcohol Vaporizer.

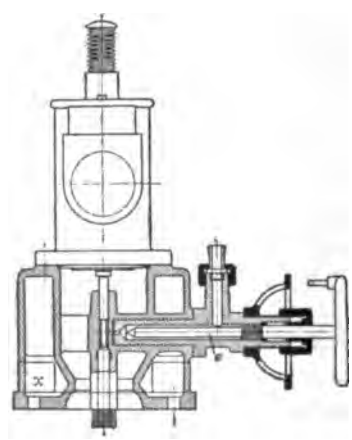


FIG. 8-24. — Dresden Alcohol Vaporizer.

by the sleeve *k*. The amount of air passing outside is regulated by the openings *n n* which are controlled by the lever *l*. The air currents passing upward carry along with them some of the liquid, the mixture being heated by the exhaust gases in *a*. The perforated plate *o* tends to aid in forming a uniform mixture.

The vaporizer of the Dresdener Gasmotorenfabrik is shown in Figs. 8–23 and 8–24. In this case the warm cooling water of the engine is used for heating. It enters the water space at *x*, Fig. 8–24. On very cold days the vaporization may be assisted at the start by pouring some hot water into the funnel *a*. Air enters at *y*, 8–23. The inlet valve *b* is automatic. It may be pushed down at will at the start by pressing down on the projecting stem *c*. The downward movement of the inlet wave opens the fuel valve *d*, to which alcohol is furnished through the needle valve *e*, Fig. 8–24. Through a number of fine openings the fuel flows into the current of air and is carried along with it, the thorough mixing being assisted by the current striking the cone *g*. As will be seen from the drawing, the heating of the charge cannot be very high. In the first place only the comparatively cool jacket water of the engine is used, and secondly the mixture itself is not in the heated chamber for any length of time.

In contradistinction to the Dresden vaporizer, the Dürr, Fig. 8–25, produces a highly heated mixture. Air enters at *x* and its amount is regulated by the throttle valve *a*. The inlet valve *b* is automatic. Alcohol is supplied through the needle valve *c*, as shown, so that when *b* is closed no flow of alcohol takes place. The current of air charged with alcohol particles passes down through *d*, up the annular space *e*, and out at *y* to the cylinder. The exhaust gases enter at *z*, and by means of baffle plates

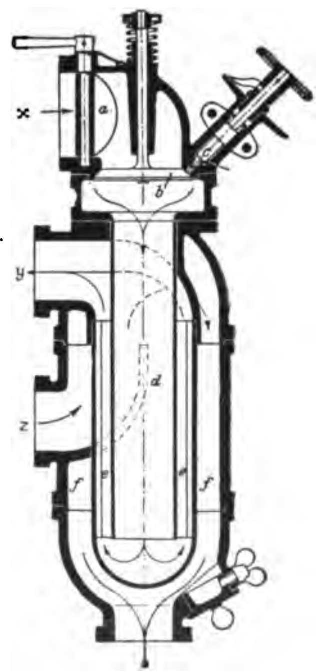


FIG. 8–25. — Dürr Alcohol Vaporizer.

are made to take the course shown by the arrows, through the space *f*. Further, the space *e* is filled with a large number of metal spirals, which connect the outside wall of *e* with its inside wall, thus furnishing a large heated surface to the passage of the charge, and facilitating the transfer of heat from the space *f* to the space *d*. Every possible way is therefore made use of to apply the heat of the exhaust gases, and this vaporizer consequently furnishes a mixture more highly superheated than that of the others.

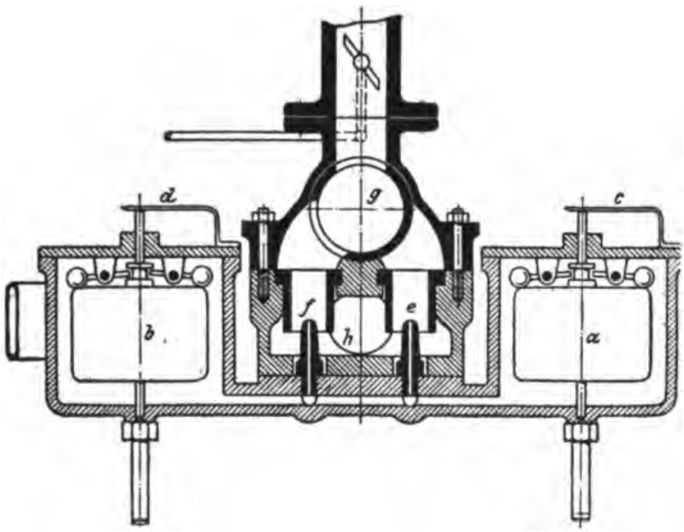


FIG. 8–26. — Gasoline-Alcohol Vaporizer.

Finally, Fig. 8–26 shows what may be called a double float carbureter, which is the form that alcohol vaporizers are likely to take. This is used on the Marienfelde machines. Assume that the chamber *a* is used for gasoline, *b* for alcohol. The needle supply valves can be held closed by the springs *c* and *d*, as shown.

On starting with gasoline, the chamber *a* is used. Spring *c* is pushed aside so that fuel can enter, being kept at constant level by the float. The valve *g* is so set that the path is open for the air from *h* past the gasoline nozzle *e*, through *g* into the cylinder. At every suction stroke the in-rushing air is then charged with gasoline issuing in a small jet from *e*. If it is desired to change to alcohol, spring *c* is pushed into place, spring *d*

is pushed aside, and valve *g* is thrown over into the position shown in Fig. 10, all the work of a moment. The air supply to this vaporizer is pre-heated.

It is quite evident from an examination of the vaporizers above described that the final temperature of the mixture is very different in the different devices. Upon this temperature, however, depends in a great measure the only other point of difference between gasoline and alcohol engines, *i.e.*, the amount of compression. All other things being the same, that fuel mixture entering the cylinder at the highest temperature will soonest give rise to pre-ignition, or at least to pounding, under an increase in compression. High temperature of charge also effects engine capacity unfavorably. It therefore becomes important to determine approximately the *lowest* practical temperature of vaporization, and the heat necessary.

Of course the amount of heat required depends upon the amount of alcohol (and its purity) per pound or cubic foot of air. Assuming that 90 volume-per cent alcohol is used, the theoretical amount of air required for perfect combustion is 7.8 pounds. Assuming that an excess of 50 per cent of air is used, which is a desirable allowance, 1 pound of 90 per cent alcohol would require in round numbers 11.7 pounds of air. With the air temperature at 60 degrees Fahrenheit, and the atmospheric pressure 14.7 pounds per square inch, this amounts to 0.0065 pound of 90 per cent alcohol per cubic foot of dry air.

90 (volume) per cent alcohol is equivalent to 87.7 (weight) per cent, so that 1 pound of air will carry, according to the above assumed ratio of mixture,

$$0.877 \times \frac{1}{11.7} = 0.075 \text{ pound of absolute alcohol,}$$

and $$0.123 \times \frac{1}{11.7} = 0.010 \text{ pound of water.}$$

To compute the air temperature required so that it may take up the above quantities of alcohol and water vapor, we must know the relation between the temperature and the degree of saturation. Meyer in his computations used the data contained in the Physikalisch-Chemische Tabellen of Landoldt & Börnstein.

Temperature Degrees F.	Vapor Tension Inches Mercury		1 pound of air contains in saturated condition, in pounds			
			At 28.95 inches Hg. Press.		At 26.05 inches Hg. Press.	
	Alcohol Vapor	Water Vapor	Alcohol Vapor	Water Vapor	Alcohol Vapor	Water Vapor
50	0.950	0.359	0.055	0.008	0.061	0.009
59	1.283	0.500	0.075	0.011	0.084	0.013
68	1.733	0.687	0.104	0.016	0.117	0.018
77	2.325	0.925	0.144	0.022	0.162	0.025
86	3.090	1.240	0.200	0.031	0.227	0.036
104	5.270	2.162	0.390	0.063	0.450	0.072
122	8.660	3.620	0.827	0.135	1.002	0.164

For our purpose the figures of the table have been transposed into English units.

In the ordinary case the air drawn into the vaporizer is not dry, but contains a certain quantity of water. Assume that the air is at a temperature of 59 degrees and just saturated. At a pressure of 26.05 inches of mercury this would correspond to 0.013 pound of water per pound of air in its initial condition. Now in the case of the average mixture above computed, the temperature of vaporization must be high enough to vaporize an additional 0.010 pound of water, making the total 0.023 pound that the air must contain per pound. It is seen from the table that a temperature of 77 degrees is quite sufficient to do this. It is also seen that at this temperature the air may take up 0.162 pound of absolute alcohol, while the quantity in the above mixture is only 0.075 pound per pound of mixture. At a temperature of 77 degrees the mixture ready for the cylinder may therefore contain the alcohol vapor in a state of some superheat. If therefore the temperature of the walls with which the mixture comes in contact is not less than 77 degrees, no fear of condensation of alcohol vapor need be entertained. In this connection a statement in the *Engineering Record* is of interest. It is there claimed that the consumption of alcohol with the jacket water leaving at 60 degrees Fahrenheit is 100 per cent higher than with jacket water leaving near 212 degrees; *i.e.*, with cooling by

vaporization. In the light of the above facts, some such increase in the consumption is quite possible.

In order to convert the liquid alcohol into vapor, a certain quantity of heat is required. According to Regnault, this amount is, for the various temperatures given, and computed above 32 degrees Fahrenheit, as follows:

At 32° F	425.7 B. T. U. per pound
68° F	453.6 B. T. U. per pound
122° F	475.2 B. T. U. per pound
212° F	481.1 B. T. U. per pound

The specific heat of liquid alcohol is close to 0.6, so that in order to convert the quantity of 90 volume-per cent alcohol contained in the assumed mixture to alcohol vapor at 77 degrees Fahrenheit would require approximately, assuming the liquid alcohol at 60 degrees Fahrenheit, $0.075 \times [458 - (28 \times 0.6)] + [0.010 \times 1100] = 44.1$ B. T. U., where 1100 B. T. U. is assumed as the heat of vaporization of water under the existing conditions.

Now the heating value of 90 volume-per cent alcohol has been shown to be 10080 B. T. U. per pound, so that the heating value on one pound of our assumed mixture will be $0.075 \times 10080 = 756$ B. T. U. The heat of vaporization required is therefore $\frac{44.1}{756} = 5.8$ per cent of the heating value of the fuel. It can be shown that the amount of heat is easily obtainable from the exhaust gases. It can also be shown that the problem may be solved by pre-heating the air only, for, assuming that the specific heat of air at constant pressure is 0.238, we would have to pre-heat the air for the assumed mixture to $\frac{44.1}{.238} + 77 = 262$ degrees Fahrenheit, which is easily possible.

If, on the other hand, not the air but the mixture is heated, then the walls need to have a temperature only sufficiently higher than 77 degrees to transfer the required amount of heat for vaporization to the mixture in the time available. To furnish more heat than this is harmful, if anything, for it affects unfavorably both the possible degree of compression and the capacity of the machine. The cooler the mixture after formation and vaporization, the better.

CHAPTER IX

GAS-ENGINE FUELS: GAS FUELS

OUTSIDE of producer gas, which has been treated in a previous chapter, the gases used for gas engine fuel are:

1. Illuminating gas. 2. Oil gas. 3. Coke oven gas. 4. Blast furnace gas. 5. Acetylene. 6. Water gas. 7. Natural gas.

1. Illuminating Gas. — Illuminating gas is made by distilling bituminous coal in retorts. From 100 pounds of average coal are obtained about 400–450 cu. ft. of cooled gas, 50–70 lb. of coke, 4.25–4.75 lb. of tar, and 8–10 lb. of ammonia liquor. Each 100 pounds of coal also require about 20 pounds of coke for the heating of the retort.

The composition of the gas varies constantly somewhat even in the same plant. The average composition is about 45–48 per cent by volume of hydrogen, 35–38 per cent CH_4, 5–8 per cent CO, and the rest heavy hydrocarbons, oxygen, nitrogen, and carbon dioxide. The gas owes its illuminating power to the heavy hydrocarbons it contains. Its heating value, however, is not proportional to its candle power. To determine the heating value the best way is to use a calorimeter. It may, however, with sufficient accuracy be computed from the analysis of the gas. Varying somewhat in this same locality, the average lower heating value is probably not far from 600 B. T. U. per cubic foot. Its density averages about .4, air being 1.0, its average weight per cubic foot, therefore, being not far from .032 lb.

The following table shows a few typical analyses of illuminating gas.*

* Mostly from Poole, the Calorific Power of Fuels.

	H	CH_4	Hydro-carbons	CO_2	CO	O	N	B.T.U. cu. ft. Lower Value
Newton, Mass.	50.59	34.80	5.23	1.16	6.16	—	2.06	599
Cleveland	34.80	28.80	11.20	.20	10.40	.40	14.20	657
Boston..............	47.49	38.67	5.21	1.04	6.74	—	.85	651
Cincinnati	45.85	39.26	5.17	.82	4.78	.41	3.71	645
Birmingham	40.23	39.00	4.76	1.50	4.05	.36	10.10	671
Glasgow	39.18	40.26	10.00	.29	7.14	.06	3.0	830[1]
Liverpool	36.44	44.28	7.90	1 70	3.39	.19	6.10	792[1]
Hanover	46.27	37.55	3.17	.81	11.19	—	1.01	661
Paris	50.10	33.10	5.80	1.50	6.30	.50	2.70	667
Average...........	43.44	37.30	6.49	1.00	6.68	.32	5.77	686

2. **Oil Gas.** — Oil gas is made by vaporizing and superheating crude oils. It may be made by vaporizing these oils in retorts which are externally heated, as in the case of the Pintsch method, or the manufacture may be carried on as in the Lowe process, in which, as previously described, the generator is first internally heated by burning crude oil, the oil to be gasified being then sent into the heated chamber together with steam under exclusion of air.

Pintsch gas, much used for railway car illumination, contains, according to Güldner,*

17.4 per cent C_2H_4, 58.3 per cent CH_4, and 24.3 per cent H by volume.

Another gas obtained from a by-product paraffin oil showed the following composition by volume:

28.9 per cent C_2H_4, 54.9 per cent CH_4, 5.6 per cent H, 8.9 per cent CO, and 9 per cent CO_2.

Oil gas as made by the Lowe process is a water gas; the composition will therefore show much more H than is indicated in the above analysis of Pintsch gas.

Wyer † in his *Gas Producer* gives the following figures:

32 per cent H, 48 per cent CH_4, 16½ per cent C_2H_4, 3 per cent N, .5 per cent O.

[1] Values evidently too high.

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren.

† Wyer, Producer Gas and Gas Producers.

Güldner estimates that the yield of gas from 1 pound of oil in the Pintsch process is from 7–10 cu. ft. of cooled gas, about .75 lb. of coke being used in the same time for heating. This amounts to about 100 lb. of oil or about 14 gallons of oil per 1000 cu. ft. of gas, in the most favorable case, as against 9–10 gallons per 1000 cu. ft. in the Lowe process. The heating value per cubic foot of the Pintsch gas, however, is higher than that of the Lowe in the ratio of about $\frac{900}{650}$, so that on the basis of thermal efficiency the two methods of making oil gas are probably not very far apart, with the chances in favor of the Lowe system on account of the coke used for heating in the other system, which must, of course, be considered in making thermal calculations.

3. **Coke Oven Gas.** — Coke oven gas when made in the old type Bee-hive oven is fundamentally the same as illuminating gas. Compare the following analysis given by Wyer of a sample of this gas with the average analysis given for illuminating gas on page 208.

H, 50.0 per cent; CH_4, 36.0 per cent; C_2H_4, .4 per cent; N, 2 per cent; CO, 6 per cent; O, 5 per cent; CO_2, 1.5 per cent.

When made in modern by-product ovens, however, the gas yield is sometimes divided and that part of the gas used for fuel has a somewhat different composition. From a diagram published by the United Coke and Gas Company of New York,* it appears that the gas evolved during the coking of the charge in a retort is divided into two parts. The entire coking period covers nearly 25 hours. The gas evolved during approximately the first ten hours, called the surplus or rich gas, is separated from that made during the rest of the period, called fuel gas. The surplus gas is high in illuminating power and in heating value, approximately 720 B. T. U. per cubic foot. The fuel gas has an average heating value of about 560 B. T. U. per cubic foot. The figures quoted are for a medium volatile coal. The rich gas from a ton of this coal in an actual case amounted to 4300 cubic feet, which was 46 per cent of the total yield per ton, this part of the gas carrying 52 per cent of the total heat value of the gas. A ton of this coal therefore yields about 9400 cu. ft. of gas.

The same treatise above quoted gives the following gas

* The United Otto System of By-product Coke Ovens.

analysis for a coal carrying from 30–32 per cent of volatile matter:

	Illuminating or Rich Gas	Fuel Gas
Illuminants	5.8	2.8
CH_4	40.8	29.6
H	37.6	41.6
CO	5.6	6.3
CO_2	3.7	3.2
O	.4	.4
N	6.1	16.1
	100.0	100.0
B. T. U. per cu. ft. higher value	730.3	551.3

Where no illuminating gases are desired the entire gas yield is recovered together. The gas is excellent for power purposes except for the somewhat high percentages of H which render the fuel mixtures liable to pre-ignition under high compression in the cylinder.

4. **Blast Furnace Gas.** — The blast furnace is really a large gas generator, with the difference that to the charge of fuel is added the burden of ore and of flux, and that the blast is air alone without admixture of steam. Owing to the calcination of the flux, which is limestone ordinarily, and to the fact that no steam is used, the gas is high in CO_2 and contains little H, the main combustible being CO. This gas had been used a long time in hot blast stoves for blast heating and under boilers to produce steam for power purposes around the works. It was Thwaite in England and Lürman in Germany who about twenty years ago called attention to the fact that this gas, although low in heating value, could be readily burned in gas engines when suitably compressed. The credit of having carried out this idea first on a large scale belongs to the Société Cockerill of Seraing, Belgium, who about 1898 put a 150 horse-power engine using this gas into operation.

It is estimated roughly that for every ton of pig iron produced one ton of coke is required, the combustion resulting in about 5 tons of gas. Taking a furnace, therefore, with an average daily capacity of 200 tons of pig iron, the gas available per hour amounts

to about 41.6 tons or 1,000,000 cu. ft. Estimating that 500,000 cubic feet are necessary for the operation of the hot blast stoves, this leaves 500,000 cu. ft. available, which if directly used in gas engines would develop about 5000 I. H. P. Of this amount 1000–1200 horse-power are probably required for power purposes around the furnace, leaving from 3800–4000 horse-power available for other work. The same amount of gas, if used under boilers, would have resulted in only about 1200 boiler horse-power, or perhaps about 2400 horse-power total in steam engines, leaving 1200 horse-power available for other purposes.

The composition and heating value of blast furnace gas naturally varies somewhat in different furnaces, and even in the same furnace under varying accidental conditions of operation. A large number of determinations led M. Witz * to the conclusion that the average heating value of a standard cubic foot was 110 B. T. U. and that it very rarely fell below 95 B. T. U., or rose above 118 B. T. U.

The average composition of the gas, according to Ledebur, appears to be

	% by Volume	by Weight
CO	24.0	24.0
CO_2	12.0	17.0
H	2.0	.2
CH_4	2.0	.8
N	60.0	58.0

The above analysis shows no water vapor, some of which is present in the gas as it leaves the stack, and it therefore probably refers to cleaned gas. It is apparent that it is an excellent gas for internal combustion engines. Its low content of H makes it suitable for high compressions, thereby overcoming any objection that may be made regarding its low heating value and difficulty of ignition.

The most serious trouble encountered in the use of blast furnace gas is the fact that it carries more or less dust, which renders cleaning of the gas imperative before use in engines. It is also apt to carry metallic vapors, which do not, however, become

* Moteurs à Gaz et à Pétrole, p. 267.

harmful until after combustion in the cylinder. The amount of impurities carried depends altogether upon the kind of ore reduced in the furnace. In some cases it is so slight that the ordinary dust settlers combined with a scrubber of some sort are quite sufficient to reduce the amount to below the allowable limit. This is, however, the exception, and the fact that the gas must be cleaned, and thoroughly cleaned, cannot be emphasized too strongly.

It is comparatively easy to take out the coarser dust carried by the gas by appliances which have long been in use for this purpose to prepare the gas for stoves and boilers. The fine dust, however, causes more trouble, and special cleaning apparatus is necessary to reduce the amount carried.

The ordinary method of procedure is to give the gas a preliminary cleaning by allowing the coarse dust to settle. The fine dust, together with the water vapor and the metallic vapors, are then taken out by passing the gas through washers, of which there are various forms, as spray towers, centrifugal fans, etc. Coke scrubbers are not satisfactory on account of the clogging up by dust which soon takes place. A dry scrubber, filled with sawdust or shavings, is sometimes used to complete the outfit of cleaning apparatus. The amount of water used in the washers varies with different types. It may be from 5–50 gallons per 1000 cu. ft. of gas cleaned, depending upon the efficiency of the apparatus. The amount of dust finally carried by the gas should not be higher than about .2 grains per cubic foot.

5. **Acetylene.** — It is only in recent years that the means for making acetylene gas in any quantity were found. To-day calcium carbide is made in quantity in the electric furnace. The gas is produced by decomposing this carbide by means of water, as per following reaction:

$$CaC_2 + 2\,H_2O = C_2H_2 + CaO{\cdot}H_2O$$

The generators employed usually regulate the amount of water supplied to the carbide receptacle. The gas is led to a holder, which by the position of its bell regulates the water supply. The amount of gas produced per pound of carbide should be theoretically 5.45 cu. ft. of dry acetylene gas. Owing to impurities, how-

ever, this is usually reduced to about 4.8 cu. ft. Combustion of this gas takes place according to the formula

$$C_2H_2 + 5O = H_2O + 2CO_2$$

Its heating value is 20673 B. T. U. per pound, or 1499 B. T. U. per cubic foot lower value.

The gas is distinguished by low temperature of ignition, lower than that of H, approximately 865 degrees Fahrenheit, high velocity of flame propagation at the best ratio of air to gas, about 12 to 1, and high maximum temperature of explosion owing to the high heating value. The first of these, low-ignition temperature, leads to pre-ignition and requires the use of comparatively low compression pressures.

6. Water Gas. — The theory of the production of water gas has been already outlined in a previous chapter. The average composition of the gas may be taken to approximate by volume 42 per cent CO, 44.5 per cent H, 3.5 per cent CO_2, the rest being O and N.

In medium sized well-handled generators each pound of coke will yield about 32 cu. ft. of gas, each pound of good anthracite coal from 24–30 cu. ft. The average lower heating value of the gas may be taken at 290 B. T. U. per cu. ft.

7. Natural Gas. — Natural gas is found in many parts of the world. It has, however, perhaps received the most extended use as a fuel for power in the United States. It is there found in New York, Pennsylvania, Ohio, Indiana, West Virginia, Kentucky, Tennessee, Colorado and California. This gas is not of constant chemical composition in the different wells, and not constant even in the same well. Marsh gas, CH_4, however, is nearly always the main constituent. According to Poole, the Ohio and Indiana fields yield a gas of the most constant composition. The following is the composition at Findlay, Ohio, and is typical of the field.

H	CH_4	C_2H_4	O	CO	CO_2	N	H_2S
1.64	93.35	.35	.39	.41	.25	3.41	.20 per cent by vol.

The above composition, however, is sometimes radically changed. Thus a gas well near Pittsburg changed the composi-

tion of the gas in three months from 9.64 per cent H to 35.92 per cent H, mostly at the expense of Marsh gas.

The heating value of this gas is high, the above Findlay gas showing a lower heating value of 962 B. T. U. per cubic foot as computed. It is a good fuel for gas engines, as it is cheap and not very liable to pre-ignition when the hydrogen is low. It is, however, of decreasing importance on account of the gradual failure of the supply.

The following table, and Fig. 9-1, give a recapitulation of the most important data for the fuel gases most often found in gas engine practice. It is to be remembered that the figures given represent approximate average results only, but for rapid calculation they are sufficiently accurate.

AVERAGE APPROXIMATE DATA FOR FUEL GASES

No.	Kind of Gas	Wt. per cu. ft. Standard in pounds	Density Air = 1	Lower Heating Value per cu. ft. B. T. U.	Least air required for Combustion cu. ft. per cu. ft.
1	Illuminating gas	.032	.40	565	5.25
2	Natural gas	.045	.55	950	9.10
3	Blue water gas	.057	.71	290	2.45
4	Oil gas, Pintsch	.056	.70	1000*.	9.50
5	Oil gas, Lowe	.040	.49	650	7.75
6	Coke oven gas	.029	.36	545	5.00
7	Producer gas from coke	.075	.93	135	1.00
8	Producer gas from anthracite	.065	.80	145	1.15
9	Producer gas from soft coal	.073	.90	145	1.25
10	Blast furnace gas	.079	.98	100	.70

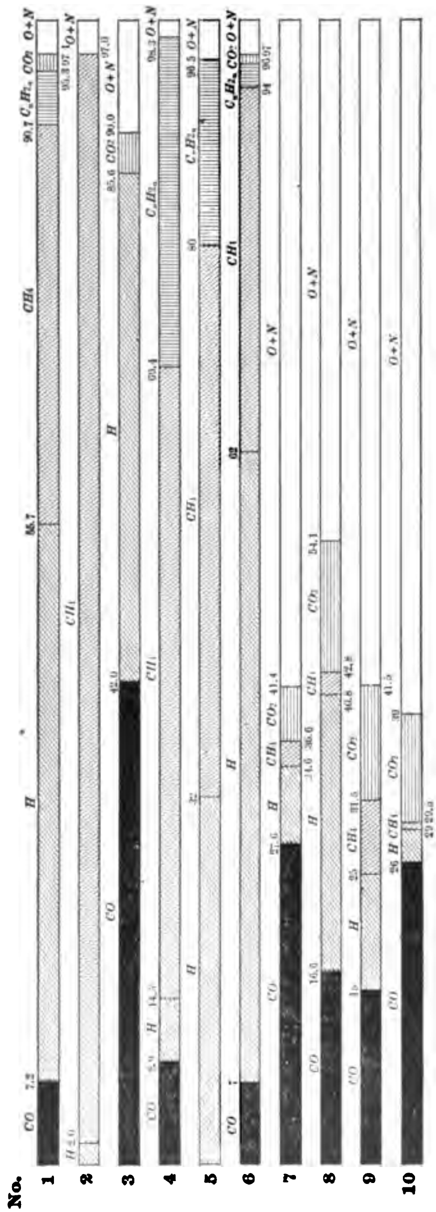


FIG. 9-1.—Approximate Composition of Fuel Gases. For Key to the different lines, see Table, p. 213. Numbers indicate per cent by volume.

CHAPTER X

THE FUEL MIXTURE, EXPLOSIBILITY, PRESSURE, TEMPERATURE

1. **Explosibility.** — When we mix a combustible gas or vapor with air there result explosive mixtures if certain ratios of air to combustible are used in each case. For each of these various mixtures the products of combustion attain certain pressures and temperatures under the same conditions. It is also found that the time interval between ignition and attainment of highest pressure varies with the ratio of mixing, *i.e.*, that the velocity of flame propagation differs.

That mixture in which just enough oxygen is present to complete the combustion of the charge of gas or of vapor shows the highest explosive pressures and temperatures, and also very nearly the highest velocity of flame propagation. Queer, and as yet not explained, is the fact that according to Clerk the highest velocity of propagation is found when the gas is a little in excess of the theoretical ratio in the mixture.

As the proportion of oxygen or of air is decreased from or increased beyond the theoretical amount for complete combustion, the resulting maximum pressures and temperatures are not so high, the explosion occurs more and more slowly, until, at the outside limits of explosibility, it approximates slow combustion.

There is therefore a range of mixtures for each gas and vapor in which range the mixture is explosive. When there are constituents present other than those resulting from mixtures of gas or vapor and air, such as burned gases, the results are again modified.

The following table shows the volume of air required per cubic foot of various gases, under standard conditions, for complete combustion, *i.e.*, for what might be called the true explosive mixture. To facilitate computation to a weight basis, a column giving the density of the gases or vapors with air = 1 = 0.08072

pounds per cubic foot standard is added. The table also shows average heating values, and finally the heating value of the true explosive mixture. It will be noted that the heating values of the mixtures show much less difference than do the gases themselves.

	Vol. of Air for true explosive mixture cu. ft.	Density of gas Air = 1 = .08072 lbs. per cu. ft.	Av. lower heating value of gas B. T. U. per cu. ft.	Av. lower heating value of true explosive mixture B. T. U. per cu. ft.
CO	2.35	.967	342	102
H	2.35	.069	297	89
CH_4	9.60	.554	952	90
C_2H_2	11.75	.915	1499	118
C_2H_4	14.10	.974	1564	103
Av. natural gas	9.00	—	880	88
Av. illuminating gas	5.25	.40	560	90
Av. water gas	2.30	.72	290	87
Av. producer gas from coke	1.00	.93	135	68
Av. producer gas from anthracite	1.15	.80	145	68
Coke-oven gas	5.00	.36	545	91
Blast-furnace gas	.70	.98	100	59

To give volume ratios for the liquid fuels is not so easy, because the volume of vapor obtained from a certain weight of a liquid fuel is not constant but depends upon the temperature. Experiments along this line are not numerous. Those made indicate that for light and medium heavy oils there is a 200-fold increase in the original volume at ordinary temperatures, while the increase is about 400 fold at the ordinary vaporizer temperature. It appears also that for heavy oils a temperature exceeding 1400 degrees is required to anywhere near complete vaporization. This explains in part the difficulty encountered in the use of crude oil in vaporizers. With the above increase in volumes the ratio of air to oil vapor by volume probably is in most cases between 25 and 30.

It is much more usual, however, to calculate the air-liquid fuel ratios by weight. This ratio for the petroleum oils is approxi-

mately 15, and for alcohol about 9, for the true explosive mixture. To get some idea of the heating value per cubic foot of the explosive mixture the volume added to the air by the vapor is sometimes neglected. This gives results which are close enough for most purposes. In the following table, however, account has been taken of the vapor volume by assuming a 300-fold increase in the volume of the liquid due to vaporization:

Liquid Fuel	Weight of air required per lb. of fuel for true explosive mixture	Av. Sp. Gr. at 60° F $H_2O = 1.0$	Average lower heating value per lb. of fuel B. T. U.	Average lower heating value per cu. ft. of true explosive mixture B. T. U.
Heavy Pa. crude		.886	19210	99.2
Light Pa. crude		.826	17930	92.0
Heavy W. Va. crude	Approximately	.873	18320	94.6
Light W. Va. crude		.841	18400	95.0
Kerosene	15.0 lb.	.80	18520	95.8
74°		.69		
Gasoline			19000	97.7
69°		.71		
Benzol, C_6H_6	13.4	.866	17190	99.3
Alcohol, 100 per cent	8.6	.794	11664	103.0
Alcohol, 90 per cent	7.8	.815	10080	104.0

To make sure of complete combustion of the charge, which is one of the primary objects, it is not usual, however, to work with the theoretic air supplies above computed, mainly because a perfect mixture is hardly ever obtained. An excess of air is therefore employed in nearly all cases. This acts beneficially in several other ways. The maximum explosion temperatures are reduced, and pre-ignition is rendered less likely. Especially is this noticeable with a gas carrying much hydrogen or with acetylene. The only limit set to the amount of excess air that can be used is the limit of the explosive range of the particular gas or vapor. But this range is in most cases so wide that in practice the limit is rarely reached. Under full load an excess of air of from 30–40 per cent is usual, and for the rich mixtures even more may be used. This excess decreases the heating value of the

mixture correspondingly below those computed above for the true explosive mixtures.

The amount of excess air any given gas can carry and still form an explosive mixture varies with the kind of gas. Experiments have been made not only on pure gas-air mixtures* under ordinary conditions of pressure and of temperature, but also on mixtures to which had been added inert gases, as CO_2, to test the effect burned gases would have upon a fresh charge. The following table shows the upper and lower limits of explosibility for some of the most common gases, together with the theoretical ratio. It is hardly necessary to point out that the real lower limit of operation, as far as economy is concerned, is the theoretical ratio:

Volume of air per unit volume of gas for	CO	H	C_2H_2	CH_4	C_6H_6	Ave. Ill. gas	Gasoline
Upper explosive limit....	5.1	9.6	28.6	15.4	36.7	11.7	40.7
Theoretical ratio	2.35	2.35	11.75	9.60	25–30	5.25	25–30
Lower explosive limit....	.33	.51	.91	6.81	14.38	4.24	19.41

Dr. Eitner's results have been graphically represented in Fig. 10–1 by F. E. Junge and published in *Power* for August, 1906. From this diagram it can be seen at a glance how much wider the explosive range is for some fuels than for others. Thus, for instance, one cubic foot of mixture will be explosive under ordinary conditions of pressure and temperature when it contains anywhere between 16 and 74 per cent of CO, a range of 58 per cent. The range for illuminating gas is 11.2 per cent, and for gasoline only 2.5 per cent.

Regarding the experiments with admixtures of inert gases, it can be said in general that the result is a narrowing of the explosive range, *i.e.*, the effect is harmful. Raising the temperatures of mixtures thus contaminated has a noticeably beneficial effect in the case of hydrogen and illuminating gas mixtures only. Regarding the effect upon the explosive limits of increasing the initial pressure of the mixture, nothing definite is known.

Carbon monoxide and hydrogen are the two most important

* Professor Eitner, in the Journal für Gasbeleuchtung, 1902.

constituents found in our power gases, and the question as to the most desirable amount of each of these in the gas has often been raised. This point is taken up by F. E. Junge * as follows:

"The temperature at which hydrogen ignites is considerably below that of carbon monoxide, and the rapidity of flame propaga-

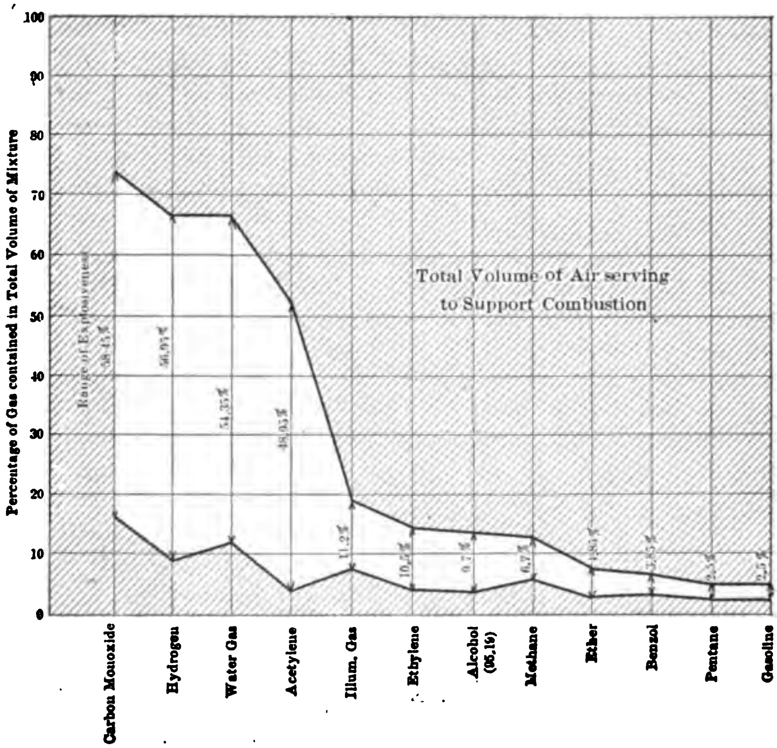


FIG. 10–1.

tion at atmospheric pressure is about 30 times greater. Its diffiusion properties are by far more favorable than those of carbon monoxide, so that its admixture with air is accomplished in a much shorter time. Its presence, therefore, determines the manner of ignition and the temperatures prevailing at various points of the inner walls. On the other hand, minor variations

* Power, August, 1906.

in the hydrogen content are of great influence on the speed of ignition or rather of flame propagation throughout the whole mixture, and therefore on engine output and consumption.

"A gas to be of ideal composition for the engine builder must, therefore, not contain too much hydrogen, so as not to make the engine over-sensitive to premature explosions, but enough so as to assist the slow and after-burning carbon monoxide, and to accelerate flame propagation throughout the mixture."

2. Pressure and Temperature after Combustion. — It has already been mentioned that the pressures and temperatures realized in explosive combustion are not as high as those calculated by the ordinary method of assuming specific heat constant. The determination of the causes of this phenomenon has always been a favorite subject for experimentation with investigators, and this is fully warranted by its importance to the theory of internal combustion engines. It will hardly be necessary to review the earlier efforts in this field. For them the reader is referred to the works of Clerk *, Grover †, Robinson ‡, Witz,§ and others. The subject has already been briefly discussed in Chapter IV of this book under the head of "Combustion and Expansion Strokes." The entire question seems on analysis to narrow down *to variation of specific heat with temperature and to after-burning*. Dissociation as one of the causes tending to produce the effects mentioned is now regarded by most writers as possible but improbable.

The most important work of recent years along the lines under discussion has been done by Langen ‖ and by Clerk.¶ It would seem that these two investigations, combined with the earlier one of Mallard and Le Chatelier, should lead to some fairly definite conclusions. They will therefore be reviewed in some greater detail.

Langen in his experiments used a cast-steel spherical vessel, about 15¾ inches in diameter. This was furnished with the neces-

* Clerk, The Gas and Oil Engine.

† Grover, Modern Gas and Oil Engines.

‡ Robinson, Gas, Oil and Air Engines.

§ Witz, Moteurs à Gaz et à Pétrole.

‖ A. Langen, Zeitschrift d. V. d. I., Vol. 47, p. 622.

¶ D. Clerk, Proc. of the Royal Society, A, Vol. 77, 1906.

sary connections for exhaust pump, gas supply cylinders, indicator gages, etc. The igniter reached to the center of the sphere. The vessel itself was surrounded by a water bath with thermometers at inlet and outlet, so that the temperature of the body of gas in the vessel could be accurately determined. The indicator was of the ordinary type except that the oscillating motion of the drum was changed to continuous motion. The details of the entire construction were very ingenious. The pressing of a single button sufficed, by electrical means, to first press the pencil against the drum, and immediately afterward to fire the charge. It was only necessary to break the current when the decrease in pressure, due to cooling of the burned gases, had become so small in two revolutions of the drum that the lines interfered.

The method of test was to exhaust the vessel and then to fill it with air a number of times until it was fair to assume that all burned gases from a previous explosion had been replaced. The vessel was then again exhausted to such a degree that, by filling with combustible gas and inert gases of the kind desired, the vessel would, at atmospheric pressure, be filled with combustible and inert gases in the proportion required. After enough time had been given for diffusion and the thermometer showed that constant temperature had been reached, the charge was fired and the diagram taken.

In order to obtain a common basis for comparison with the work of previous experimenters, Langen recomputed all of his results and those of Bunsen, Berthelot and Vieille, and Mallard and Le Chatelier, on the assumption that the temperature at the moment of explosion was 0 degrees Centigrade. Figs. 10–2 and 10–3 * show graphically the results of all of these experiments, the first for CO, the second for H as the fuel gas. In the diagrams π_0 stands for the ratio of explosion pressure observed to the pressure before ignition if the temperature of the fuel mixture is 0 degrees Centigrade at the start. m is the ratio of the volume of inert gases compared with the volume of fuel gas. These inert gases were N, O, H or CO, or any mixtures of these four as indicated. The results marked e were obtained for air-fuel gas mixtures.

* Zeitschrift d. V.d. I., 1903, p. 623.

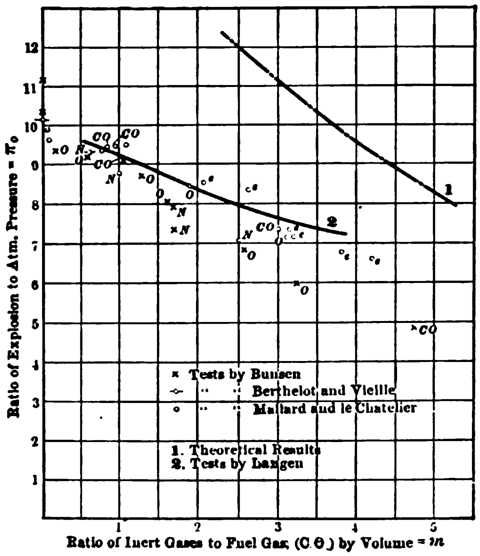


FIG. 10–2.

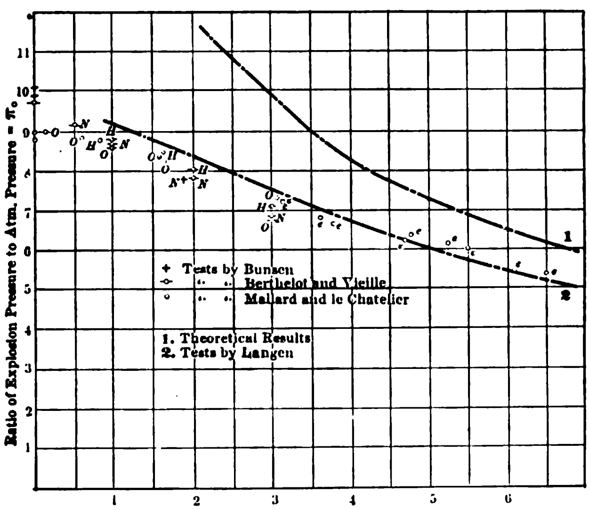


FIG. 10–3.

Langen, in analyzing the results shown in the above diagrams, makes the following observations:

1. The computation of explosion pressures on the assumption of constant specific heat and complete combustion furnishes values which considerably exceed those actually observed.

2. For equal ratios of inert diatomic gases to fuel gases, the *kind* of inert gas used seems to have no influence upon the explosion-pressure, *as far as the same observer is concerned.* This would lead to the conclusion that the molecular heats of the so-called simple or diatomic gases are equal to each other, at least up to 4500 degrees Fahrenheit.

Regarding the results of his own experiments, Langen is of the opinion that in fuel mixtures containing CO, dissociation of CO_2 sets in when the temperature exceeds 1900 degrees Centigrade (3450 degrees Fahrenheit). He bases his opinion on the abnormal position of the cooling curve as observed on the diagrams taken for such mixtures. And since the amount of this dissociation is indeterminate, no definite equation expressing the relation of maximum pressure to initial pressure can be established, at least for temperatures exceeding 3400. For hydrogen mixtures, on the other hand, the cooling curves on the diagrams are always normal. Hence the dissociation limit for steam does not seem to have been reached even with the strongest mixtures.

From that part of his experiments for which complete combustion can be assumed, Langen derives equations for mean molecular heat of diatomic gases, and for carbon dioxide and steam. The temperature limits for the field so covered are not very wide, 1500–1700 degrees Centigrade (2730–3100 degrees Fahrenheit), and further it was assumed that the molecular heat is a linear function of the temperature. Transposed to read mean specific instead of mean molecular heats, these equations are as follows:

$$\begin{aligned}
&\text{For } CO_2, && C_v = .152 + .0000591\,t.\\
&\text{For } H_2O, && C_v = .328 + .000119\,t.\\
&\text{For } N, && C_v = .171 + .0000215\,t.\\
&\text{For } O, && C_v = .150 + .0000188\,t.
\end{aligned}$$

where t is in degrees Centigrade.

The formulæ of Mallard and Le Chatelier agree with the

above as regards the diatomic gases, O and N. For CO_2 and H_2O these observers obtained results which gave the following relations:

$$\text{For } CO_2, \ C_v = .143 + .0000834\,t.$$
$$\text{For } H_2O, \ C_v = .312 + .000182\,t.$$

These formulæ show a somewhat more rapid increase of C_v with temperature than do those of Langen. Langen observes in explanation of this discrepancy that Mallard and Le Chatelier's formula for CO_2 is obtained from results for which the temperatures were from 1700 to 2000 degrees Centigrade, and that the formula for H_2O is based on experiments for which the temperatures were very much higher than for his experiments. In the former case dissociation was shown to be more than likely, in the latter the formula gives results which do not seem to apply very closely for the important temperature range between 2250 and 4000 degrees Fahrenheit. It is plain, therefore, that Langen's formulæ promise greater accuracy.

The second important investigation in this field was made by Clerk, and by him reported to the Royal Society. His method of operation is so decidedly different from that of Langen and the earlier experimenters that it becomes both interesting and important to see how far his results agree with those already mentioned.

The method of experiment is best described in Clerk's own words:

"It consists in subjecting the whole of the highly heated products of the combustion of a gaseous charge to alternate compression and expansion within the entire cylinder while cooling proceeds, and observing by the indicator the successive pressure and temperature-falls from revolution to revolution, together with the temperature and pressure rise and fall due to alternate compression and expansion. The engine is set to run at any given speed, and at the desired moment after the charge of gas and air has been drawn in, compressed, and ignited, the exhaust valve and charge inlet valves are prevented from opening, so that when the piston reaches the termination of its power stroke, the exhaust gases are retained within the cylinder, and the piston compresses them to the minimum volume, expands them again to

the maximum volume, and so compresses and expands during the desired number of strokes."

To attempt to explain the method of evaluating the expansion and compression lines so obtained would lead too far for the scope of this book. The reader is referred to the original article.*

The engine operated with coal gas. The average composition of the working fluid, as calculated from the analysis of the gas, was H_2O (assumed gaseous), 11.9 per cent by volume; CO_2, 5.2 per cent; O, 7.9 per cent, and N, 75 per cent. The mixture as actually used varied somewhat from this composition, but since the percentage of N is nearly constant, this variation can have but small effect upon any specific heat calculations.

For this mixture, Mr. Clerk, on the basis of his experiments, found the following mean specific heats, expressed in foot pounds per cubic foot of working fluid at 760 mm. and 0° C.

Range of Temperature °C	°F	Mean specific heat in ft. lbs. per cu. ft. at 760 mm and 0° C.
0– 100	32– 212	20.3
0– 200	32– 392	20.9
0– 400	32– 752	21.9
0– 600	32–1132	22.8
0– 800	32–1472	23–6
0–1000	32–1832	24–6
0–1200	32–2192	24.6
0–1400	32–2552	25.0
0–1500	32–2732	25.2

Now to compare these results with those of Mallard and Le Chatelier and of Langen, the easiest way would be to reduce them to the ordinary specific heat basis, and then to compute a series of specific heats for the same temperature ranges and for the same mixture as used by Clerk by the aid of Mallard's and of Langen's formulæ. This involves the assumption that these formulæ hold for the lower temperature ranges. The following table shows the figures so obtained, and Fig. 10–4 gives a graphical comparison.

* See foot-note, page 221.

MEAN SPECIFIC HEAT FOR MIXTURE CONTAINING BY VOLUME 11.9 % H_2O, 5.2 % CO_2, 7.9 % O, and 75 % N.

Temperature Range °C	°F	Mallard & Le Chatelier	Langen	Clerk
0– 100	32– 212	.1805	.1826	.1854
0– 200	32– 392	.1843	.1858	.1910
0– 400	32– 752	.1930	.1922	.2000
0– 600	32–1132	.2006	.1985	.2083
0– 800	32–1472	.2083	.2047	.2157
0–1000	32–1832	.2161	.2112	.2202
0–1200	32–2192	.2238	.2176	.2248
0–1400	32–2552	.2315	.2239	.2284
0–1500	32–2732	.2355	.2271	.2303

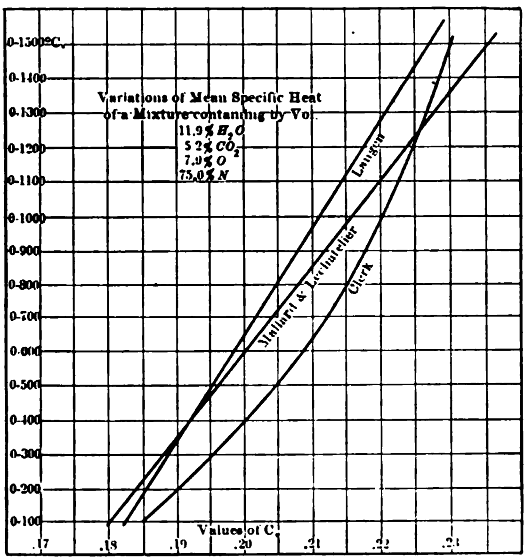


FIG. 10–4.

It is plain from Fig. 10–4 that the question of the variation of specific heat with temperature cannot be considered entirely solved. It is true that Stevens * made experiments on air which check the results of both Mallard and Le Chatelier and of Langen

* Ann. d. Phys., 1902.

very closely. It is therefore fair to assume that the formulæ derived for diatomic gases are correct. The discrepancies observed between the results plotted in Fig. 10–4 are therefore with strong probability due to error in the available data for H_2O and CO_2. Further work is therefore required, and a check of Clerk's results is especially desirable.

3. Velocity of Flame Propagation and Time of Explosion. — Experiments on the velocity of flame propagation in a given mixture, like the experiments on specific heat, have not led to any definite result. And although definite information on this point is desirable, on account of the connection between velocity of flame propagation and possible maximum engine speed, in any given case there are so many factors affecting the problem in actual practice, that the application of the results of laboratory experiments to actual conditions is of doubtful value. Thus the velocity with which the flame spreads through a mixture depends upon the kind of fuel, the composition and purity of the mixture, its temperature and pressure, and upon the location of the igniter and the shape of the combustion chamber. Further than this, the degree of mechanical agitation in the mixture at the moment of explosion has a marked influence upon the velocity.

If we ignite a mixture in a tube, closed at one end, from the closed end, the pressure generated seems to project the flame ahead of the pressure wave toward the open end of the tube with a much higher flame velocity than would have been observed if the ignition had taken place from the open end. The same effect should be observed in a tube closed at both ends. Ignition from the open end gives the true velocity, as the flame then spreads by contact only. In an actual engine cylinder, with the volume increasing with the movement of the piston, we may expect the velocity of propagation to be somewhere between those found for ignition from the open end of a tube and those for ignition in a closed tube.

Mallard and Le Chatelier used the open-tube method, measuring the time interval between the passage of the pressure wave between two points on the tube. Their results, as given by Clerk, for hydrogen were as follows:

Mixture	Velocity of Pressure Propagation ft. per second
1 vol. H + 4 vols. air	6.56
1 vol. H + 3 vols. air	9.20
1 vol. H + 2⅓ vols. air	11.10
1 vol. H + 1⁶⁄₇ vols.air	12.40
1 vol. H + 1½ vols. air.	14.30
1 vol. H + 1 vol. air	12.30
1 vol. H + ⅓ vol. air	7.55

The true explosive mixture for hydrogen and air is 1 vol. of H to 2½ vols. of air. It might be supposed that this would be the mixture showing highest velocity of propagation. For some unexplained reason a certain excess of hydrogen shows the highest

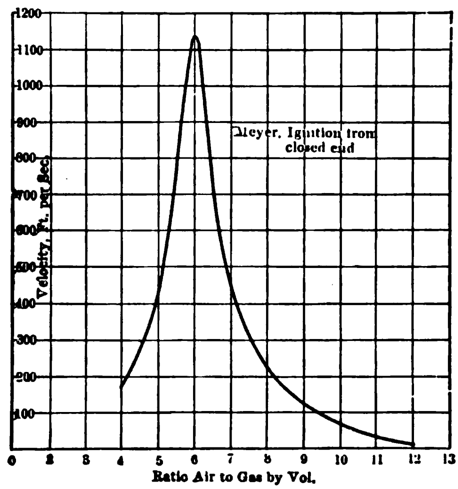


FIG. 10–5.

result. A similar phenomenon was observed with coal gas and air mixtures by these observers.

Meyer * used an apparatus very similar to that of Mallard and Le Chatelier, except that very sensitive platinum thermometers were employed to measure the passage of the flame, instead of diaphragms to measure the passage of the pressure wave. The fuel employed was coal gas. All tests were made with the mix-

* E. J. Meyer, Sibley College thesis, 1905.

ture at atmospheric pressure at the time of ignition. Figs. 10–5 and 10–6 show the results graphically, the former being obtained for ignition from the closed end, the latter for ignition from the open end. The much higher velocity of the first curve bears out the statement previously made. The fact that the maximum velocity occurs with the same gas-air ratio shows that the true rate of inflammability has not been changed, but that mechanical actions alone are responsible for the difference in the observed results.

Closely connected with the velocity of flame propagation, and subject to the influence of accidental accompanying conditions to the same degree, is the time of explosion. The most extensive work in this field was done by Clerk. The apparatus employed by him was very similar to that used by Langen in his specific

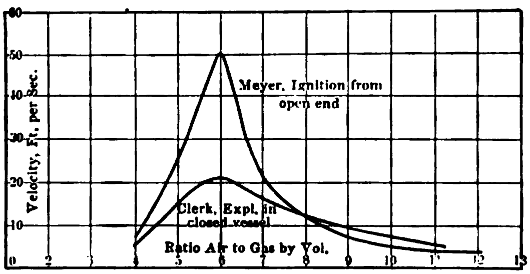


Fig. 10–6.

heat experiments. The fuel mixture was ignited at constant volume and a pressure diagram obtained on the rotating drum of an indicator. Unfortunately the experiments were confined to coal gas, a few figures only being obtained for hydrogen and none for the power gases so important to-day. The figures found for hydrogen were as follows, the time of explosion being the time interval from the moment of ignition to the attainment of maximum pressure.*

Mixture by Volume		Time of Explosion in seconds
Air	Hydrogen	
6	1	.150
4	1	.026
2.5	1	.010

* Clerk, The Gas and Oil Engine, p. 101.

Clerk's results for mixtures of air and Oldham coal gas are represented in Fig. 10–7. The volume ration showing fastest time of explosion, *i.e.*, 6 to 1, agrees with the mixture for which Meyer found the greatest velocity of flame propagation. This mixture also showed about the highest pressure development in Clerk's experiments, 90 pounds per square inch.

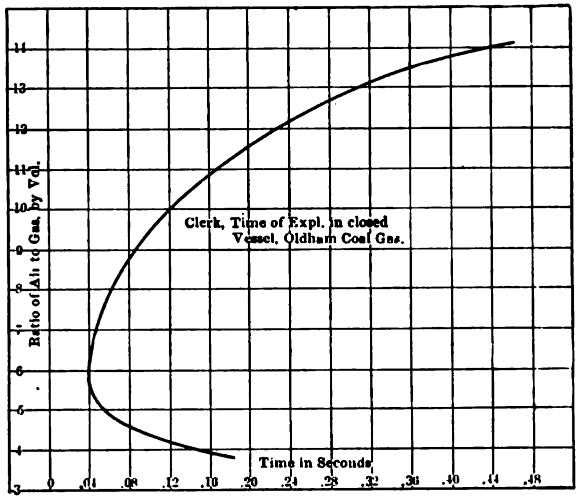


FIG. 10–7.

In Clerk's experiments the pressure at the moment of ignition was atmospheric in each case. Koerting * carried on similar experiments with coal gas but used compressed mixtures, although the pressures used were low. The summary of his results is as follows:

Mixture by Vol.		Pressure before Ignition Lbs.	Time of Explosion seconds	Velocity of Propagation ft. per sec.
Air	Gas			
7.5	1	15.0	.032	23.0
		37.0	.036	20.4
5.42	1	15	.01	74.0
		37	.0125	59.0

* Koerting, Zeitschrift d. V. d. I.

This table shows that compressing the mixture retards the velocity of flame propagation, but that the amount of retardation is less in lean than in rich mixtures of the same fuel.

Koerting's figures do not agree well with those of Clerk, although they were obtained with similar apparatus. From Fig. 10–7 the time of explosion of a 7.5 to 1 mixture would have been about .056 seconds according to Clerk, as against .032 seconds found by Koerting. This is too great a difference even assuming a considerable difference in the composition of the fuel. The length of Clerk's vessel up to the indicator piston was about 10 inches, which, with a time explosion of .056 seconds, gives a velocity of propagation in a closed vessel of 14.8 feet per second as against 23 feet found by Koerting.

For the purpose of comparing Clerk's results with those of Meyer on flame propagation, the times of explosion as shown by Fig. 10–7 have been transposed to the basis of velocity in feet per second. The resulting curve has been drawn in on Fig. 10–5 with Meyer's results. It is seen that the velocity of propagation in a closed tube according to Clerk is lower than that found by Meyer for ignition from the open end of a tube, except for ratios exceeding 8 to 1, and here the difference is inconsiderable. This is contrary to what might be expected, because, as before explained, and also mentioned by Clerk, if explosion takes place at constant volume in a closed vessel, the part of the mixture first ignited instantly expands and shoots the flame into the rest of the mass, thus increasing the velocity of propagation. As compared with Meyer's results for ignition from the closed end of a tube, Clerk's results are very much lower, in fact only about $\frac{1}{60}$ at the best ratio for the gas. Koerting's figures, on the other hand, slightly exceed Meyer's results for ignition from the open end.

CHAPTER XI

HISTORICAL SKETCH OF THE INTERNAL COMBUSTION ENGINE

1. **Origin obscure.** — The origin of the internal combustion engine is imperfectly known; as it exists at the present time it is the result of a long-continued development which began first with a period of speculation which, through the efforts of numerous inventors, finally resulted in a practical, operative machine. No single person can be considered as the inventor of the internal combustion engine. Its history shows the existence of three periods: (1) that of speculation and invention; (2) that of development, and (3) that of application.

2. **The Period of Speculation and Invention.** — The gas engine previous to 1860 was not a practical, commercial machine nor had it been used to any great extent for the purposes of producing power. Previous to that time various publications and patents show that nearly all of the types known at the present time had been discovered, although the records are imperfect as to the actual and practical use of such machines. It is reasonable to believe that many of the forms described or patented were actually built and operated experimentally if not commercially.

Aimé Witz gives the credit * for the first internal combustion engine to the Abbé Hautefeuille, who describes an engine in 1678 in which water is raised by utilizing the partial vacuum which results from burning gunpowder in a cylinder and cooling the gases remaining.

A similar engine was described by Huygens in a memoir entitled "Une nouvelle force mouvant par le moyen de la poudre a canon et de l'air," which appeared in 1680. Denis Papin constructed an internal combustion engine similar to that described by Huygens in 1690, but on account of imperfect workmanship

* Moteurs a Gaz et a Pétrole.

obtained results very much inferior to those produced by a steam engine and abandoned the powder engine as an impractical machine.

In the powder engines the method was a fairly practical one; a small quantity of gunpowder exploded in a large cylindrical vessel expelled the air through check valves, thus leaving, after cooling, a partial vacuum below the piston. The pressure of the atmosphere did work by moving the piston downward.

For a long time after this attempt, the internal combustion engine seemed to have been practically forgotten, as the next description of its construction was not written until after Watt had developed and improved the steam engine. Considering the low condition of the state of the mechanical arts and the difficulty of obtaining good workmen and proper materials, which Watt experienced and which he was only able to overcome by spending years of time, it can be readily understood why little or no substantial progress was made. Watt not only solved the problem relating to the method of improving the engine, but he also developed in a very great measure the art of constructing such engines and of producing proper materials for their manufacture.

The greater portion of the information relating to the design and improvement of the gas engine during the period previous to any extensive commercial use is obtained from the records of the English, French and American patent offices. These records have the advantage over other publications of being definitely dated and of concisely and accurately describing the machine and its mode of operation.

The first internal combustion engine described in the patent records was patented by Robert Street in England on the 7th of May, 1794. It consists of a motor cylinder, with a piston, the bottom of which is heated by fire. The patent shows a pump driven by a lever. The fuel is described as a small quantity of tar or turpentine which is projected onto the hot part of the cylinder so that the liquid is instantly converted into an inflammable vapor. The raising of the piston by means of a lever sucks in external air and also flame for ignition, which causes the explosion. The details as described are crude, but the principle of operation is the same as in the Lenoir engine which was patented in 1860.

Phillippe Lebon patented in France, in 1801, a gas motor in which the gas was compressed in a cylinder external to the working cylinder previous to ignition. In this patent there is described the use of an air pump for compressing atmospheric air, a gas pump for compressing gas, and the use of electricity for ignition. Lebon died September 22, 1804. Witz states "that it is probable that the industry of building gas engines would have dated at the beginning of the century as a practical commercial industry, instead of 1860, had he lived."

Various inventors early in the century proposed the use of explosive powders, of air saturated with hydrocarbon and of hydrogen gas produced by chemical means, as fuels for internal combustion engines, but these did not, so far as can be ascertained, result in any practical improvement.

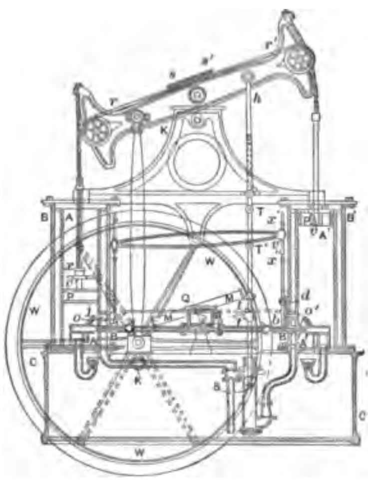


Fig. 11-1. — S. Brown, 1823-26.

In 1823 and 1826 Samuel Brown obtained English patents for an ingenious atmospheric air engine which, although very cumbersome and uneconomical, was applied to practical uses. The engine, Fig. 11-1, was operated by burning the combustible in a vessel adjacent to the working cylinder, which resulted in expelling a portion of the air it contained. A jet of water was then thrown in which lowered the temperature and by so doing produced a vacuum. Motion was produced by the atmospheric pressure acting alternately on the sides of the piston in the working cylinder, which was arranged adjacent to the vacuum-producing chamber and put in alternate connection with it by means of a proper valve motion. This engine, although referred to by Clerk "as being the first gas engine undoubtedly put at work," is an external combustion gas engine somewhat similar to the Wilcox engine already described, except that there was vacuum rather than pressure in the combustion vessel. According to the

Mechanics Magazine, published in London, a boat was fitted with a Brown engine and ran experimentally upon the Thames. Another engine was made in combination with a road carriage (see *Mechanics Magazine*, December 24, 1825); this is also referred to in a Report of a Committee of the House of Commons, reprinted as a public document in the United States in 1832.

W. L. Wright obtained an English patent (No. 6525 of 1833) for an internal combustion engine in England. This patent is accompanied with elaborate drawings giving in detail the proposed construction. The engine, Fig. 11–2, is shown as double acting, the piston receiving two impulses for every revolution of the crank shaft. It is also shown as provided with pumps adapted to compress the air and gas a few pounds above the atmosphere previous to their introduction into the cylinder. The engine was provided with a fly-ball governor for controlling the quantity of gas and air as required to produce uniform speed. The charge is ignited when the working piston is at the end of its stroke by an external flame, burning in air, which is connected with the charge at the proper time by a valve opened by the mechanism of the engine. The charge was not under sensible compression at the time of ignition. The engine as shown has water-jacketed cylinder and piston, poppet exhaust valves operated by cams, and appears well proportioned throughout. Much credit is due to the governing device shown, and it would be difficult to state why the motor did not succeed, unless it may have been due to the lack of demand for any other motor than the steam engine.

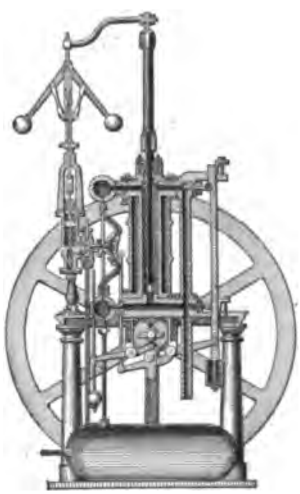


FIG. 11–2. — Wright, 1833.

James Johnson took out an English patent in 1841 for a gas engine to be operated by hydrogen gas. In this engine the piston is forced to the end of the cylinder by the explosion of the gas, after which a vacuum is formed underneath the piston and the piston is returned by atmospheric pressure. It illustrates the

expansion of the products of combustion below atmospheric pressure, a method which as yet has not met with much practical success in the operation of gas engines.

William Barnett obtained a patent in England (No. 6015, of 1838), which describes the construction and mode of operation of a two-cycle gas engine, single and double acting, in which the explosive mixture is compressed previous to ignition. It shows three forms of engines, one type in which the compression is entirely performed outside the working cylinder and which is shown as both single and double acting. It also shows another type in which the compression is completed inside the working cylinder, this latter form being almost identical with the modern two-stroke cycle engine which has already been described.

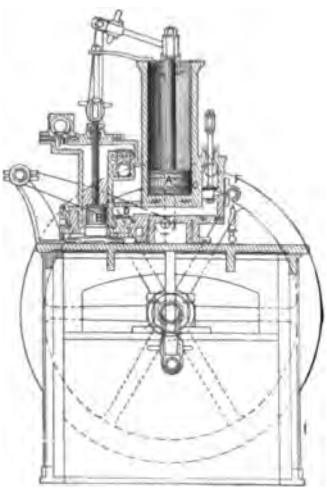


Fig. 11-3. — Barnett, 1838.

Barnett also shows an igniting device, described in Chapter XIII, which is adapted to ignite the gaseous mixture while under compression. This method of ignition is of interest as it was used in a slightly different form by Otto in the Otto engine in 1877. Barnett also describes a method of igniting by bringing the explosive mixture in contact with spongy platinum which is located in a cavity near the head of the cylinder.

Figure 11-3 is reduced from patent drawing and shows a section of the single-acting engine in which A is the motor piston. The cylinder is open at the top, B is a double-acting pump which serves to supply atmospheric air to form an explosive mixture on one side and to exhaust the products of combustion on the other; the pump for supplying fuel gas under compression stands back of the air pump and is not shown in the figure. During the ascent of the piston, A, the air and gas pumps have been drawing in air and gas, which on the descent of the pump pistons are forced into the receiver, D, which is separated from the working cylinder by the piston slide valve, E. In the meantime the piston A has discharged the exhaust gases through the valve E.

This valve closes communication with the air when *A* reaches the lower dead center and opens communication with the receiver *D*. At this instant the charge is ignited, and the gases under compression pass into the working cylinder through the slide valve *E*. The igniting device shown at *F* is operated by the motion of the valve.

The gases at the time of explosion are in the receiver *D*, and flow through the port after ignition precisely as steam would do. The pressure in the motor cylinder falls by expansion with increase in volume due to motion of piston.

Barnett's second engine is identical with his first except that it is double acting.

Barnett's third engine, shown in Fig. 11–4, is of great interest since it is the forerunner of the modern two-cycle engine. The engine shown in the figure is double acting. Like the first engine it has three cylinders, motor cylinder *A*, air pump *B*, and a gas pump not visible in the figure. The air and gas pumps are single acting but are operated by means of gearing so as to make twice as many strokes as the working piston. The ignition is performed by spongy platinum with which the gases under compression are brought in contact. The operation of the engine is as follows, supposing that when the piston is in the position shown in the drawing it is moving upward and the upper end of the cylinder is charged with air and gas under compression. When the piston has completed its up stroke, the contact of the platinum with the compressed mixture, produced by the ascent of the piston, causes explosion, which in turn impels the piston to the bottom of its stroke. During the first part of the descent and until the piston passes the port, M, at the center of the cylinder, the products of combustion below the piston are being exhausted, either into the atmospheric air or into an

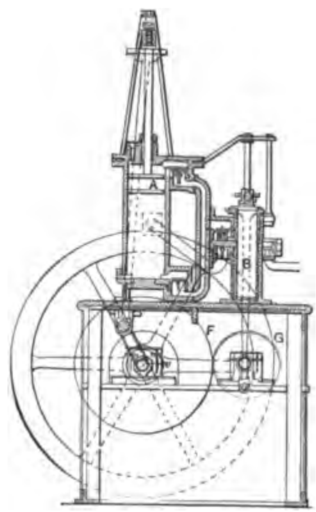


Fig. 11–4. — Barnett's Third Engine.

exhaust pump, which may be used if desired. At the same time the air and gas pumps draw in their respective charges. During the latter half of the descent of the piston the air and gas pumps are forcing the mixed air and gas into the cylinder below the piston where it is further compressed and exploded at the end of its stroke, in which case the piston is forced upwards and the operation is repeated.

Stuart Perry patented in the United States, May 25, 1844, and in Great Britain through the agency of Joseph Robinson (No. 9972 of 1843), a gas or vapor engine which was provided with means for compressing the charge previous to ignition. The method of ignition in the Perry engine was similar to that in the Wright engine. The engine was especially designed for the use of liquid hydrocarbon, and for this purpose it was provided with a carbureter through which a portion of the air on its way to the engine is forced and which may be heated by the exhaust gases. The Perry engine is double acting and provided with a rotating valve which by means of gearing can be reversed in direction, thus reversing the engine.

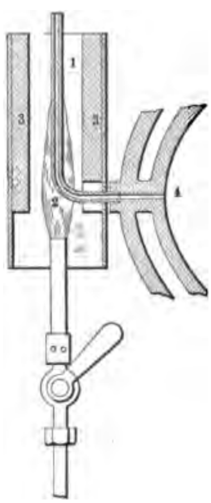


Fig. 11–5. — Newton's Hot Tube Igniter.

The Perry patent of 1844 was followed by one in 1846 which showed an engine with poppet valves and with means for igniting by a hot tube consisting of a platinum cup kept hot by a gas flame. In this latter engine the charge was under compression at the time of ignition. In the latter patent reference is made to the actual use of a gas engine with carbureting device, from which it would appear that Perry at least anticipated in design many later constructions.

A. V. Newton obtained a patent in England (No. 562 of 1855), for an ignition device which was identical with that formerly patented in America by Drake, and which is now known as the hot-tube method of ignition. The igniting arrangement consists of a cast-iron tube, closed at the outer end, which projects into a recess formed in the side of the cylinder. A gas flame, Fig. 11–5, keeps a section of this tube at a temperature sufficient to

explode the charge when the piston uncovers the opening. The engine shown in the Newton patent is a non-compression double-acting engine with water jackets.

Barsanti and Matteucci of Florence, Italy, took an English patent (No. 1655 of 1857) for a free piston vacuum engine, which is of interest from the fact that it was of essentially the same class as a successful type of engine which was introduced some years later by Otto and Langen. In this engine the force of the explosion was utilized in moving the piston when free from the connecting rod, the work being done on the return stroke by the weight of the piston and by atmospheric pressure acting on its upper side.

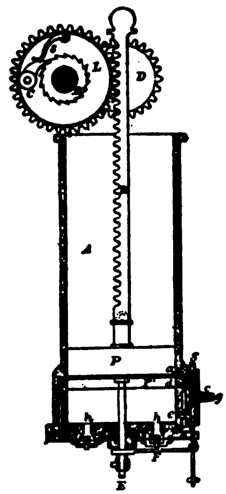


FIG. 11–6.—Barsanti and Matteucci.

In construction the cylinder of the engine is vertical, open at the top and very long, Fig. 11–6. The charge consisting of gas and air enters when the piston is drawn up a short distance and is exploded by an electric spark. The piston rod carries a rack which is in engagement with a gear wheel connected by a ratchet and pawl to the main shaft. When the piston is shot rapidly upward the gear wheel turns without moving the main shaft. When the piston returns the ratchet connects the gear wheel of the main shaft so that it is turned in the direction for producing work.

Clerk states that the method illustrated in this engine possesses three advantages: rapid expansion, a large amount of expansion, and also some of the advantages of a condenser.

3. **Period of Development.** — LENOIR. The first internal combustion engine to attain any marked degree of commercial success was patented by J. J. E. Lenoir, in France, January 4, 1860, and in the United States March 19, 1861.

The Lenoir engine was of simple construction and belonged to a type which had been previously described by several inventors. Its success was evidently due to the good proportions which characterized the design. The engine cylinder is shown

in section in Figs. 11–7 and 11–8. This engine belongs to the class in which the ignition takes places while the charge has practically constant volume and, in the case of the Lenoir engine,

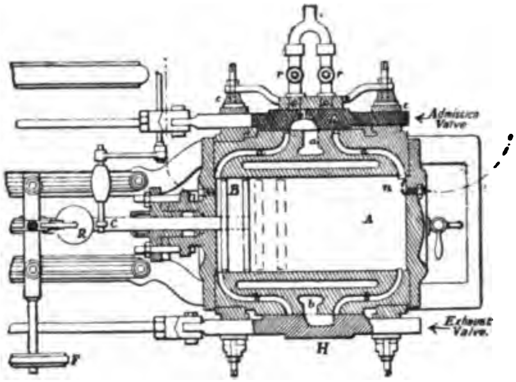


FIG. 11–7. — Lenoir Engine, 1860.

had not been compressed previous to ignition; it is devoid of any compression mechanism. In structure it resembles that of a double-acting steam engine with separate slide valves for the

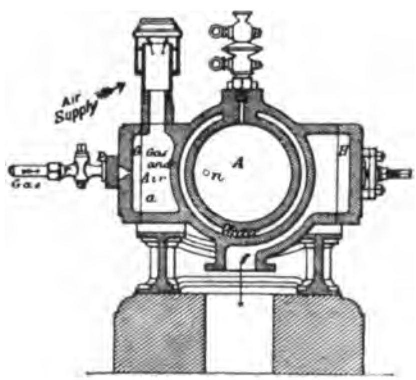


FIG. 11–8. — Lenoir Engine, 1860.

admission and exhaust. Thus in Fig. 11–7 the charge is admitted by the slide valve G, and is exhausted by the slide valve H. It is drawn into the cylinder by the partial vacuum produced by the motion of the piston, the slide valve being arranged to open the ports at the beginning of the stroke and

close them at about the center of the stroke. The charge is exploded by electrical means when the piston is at about the middle of its stroke, and produces the requisite pressure to force the piston onward and keep the engine in motion. The exhaust valve opens to discharge the products of combustion during the return stroke. The operation of both ends of the engine takes place alternately and tends to produce a uniform motion of the fly-wheel.

The engine is shown provided with cross-head, connecting rod, and governor similar to a steam engine. The governor is arranged to throttle the supply of gas as required to produce uniform speed. The engine was provided with a device for timing the electrical spark. The source of electricity was a primary battery arranged to intensify the current by an induction coil.

Although the consumption of gas by this engine was always high and the power produced in porportion to the cylinder very small, yet it possessed certain advantages. Its mechanism is simple, its explosion nearly without shock, and its action very smooth. This engine was sold in large numbers and manufactured both in France and England.

Dugald Clerk quotes from an article in the *Practical Mechanics Journal* of August, 1865, showing that from 300 to 400 engines were at that time at work in France. He also states that the Reading Iron Works Co., at Reading, England, made and delivered 100 engines.

Witz states that this engine was received with great enthusiasm and that many predicted that the last hours of the steam engine had sounded and that the star of Watt paled before that of Lenoir. This enthusiasm resulted in a marked exaggeration both as to economy and capacity, which was followed by a reaction during which time the engine was called a humbug and many were broken up for old iron. It was, however, appreciated at its full value at a later time, as it was found extremely well adapted for small establishments requiring from ½ horse-power to 4 horse-power. For such uses the consumption of gas was guaranteed to be less than 70 cu. ft. per horse-power hour and this guarantee seems to have been usually realized. Juries at the expositions of London in 1862, of Paris in 1867, of Vienna in 1873, recognized the merit of Lenoir.

The Lenoir engine is described as adapted to be operated either with gas or with the vapor of liquid hydrocarbon, and in a French patent of 1861 a carbureter is shown for mixing the vapor of oil with air.

The engine of Lenoir was used for various purposes, for instance, printing, pumping water, driving lathes, and also for the purpose of propelling a vehicle and a road carriage.

In the report of the Vienna Exhibition of 1873, R. H. Thurston quotes the following statements of M. Claudel in reference to the experiments of M. Tresca on the Lenoir engine:

"The speed of the engine is variable."

"The failure to ignite a single charge will stop it."

"To start it it is necessary to give it several revolutions by hand."

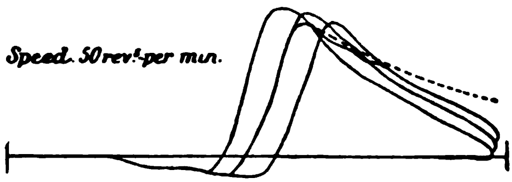


FIG. 11–9. — Diagram Lenoir Engine.

"Lubricant must be abundant and the amount of oil cannot be estimated at less than 0.5 kilogram (1.1 lbs.) per day."

"To obtain the best effect, it is necessary to open the inlet before the complete closing of the exhaust valve."

"A machine of 0.24 meter (9.5 inches) diameter of cylinder produced very nearly one horse-power."

The best performance claimed for the Lenoir engines is 70 to 74 cu. ft. of gas per horse-power per hour. M. Tresca reported a consumption of 95.28 cu. ft. A typical indicator diagram from a Lenoir engine is shown in Fig. 11–9.

HUGON. — It has been shown that Hugon had experimented with an engine of the Lenoir type two years before Lenoir.* But while the obvious disadvantages of the construction did not prevent Lenoir from putting his engine on the market, Hugon was not satisfied with the solution of the problem and turned his attention to the indirect acting engine of the type of Barsanti

* Memoirs des Ingenieurs Civils, 1860, p. 159.

and Matteucci. Only after the Lenoir engine proved in a certain manner a commercial success did Hugon return to his former engine, and, making some improvements on Lenoir, brought the engine out in 1864. In this machine, Figs. 11–10 and 11–11, electric ignition was replaced by flame ignition, which was much the surer method of igniting a charge in those early days, and

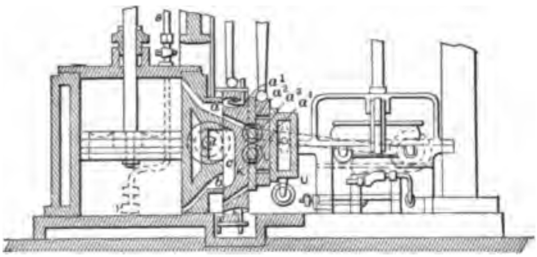


FIG. 11–10. — Cylinder, Hugon Engine, 1864.

both exhaust and inlet were operated by one valve in order to give this valve the advantage of cooling by the incoming cold mixture. To overcome the very serious defect of the Lenoir engine existing in the extremely rapid wear of the valves, Hugon reduced the temperatures of the cycle by injecting water into the charge.

The Hugon engine was somewhat superior to that of Lenoir

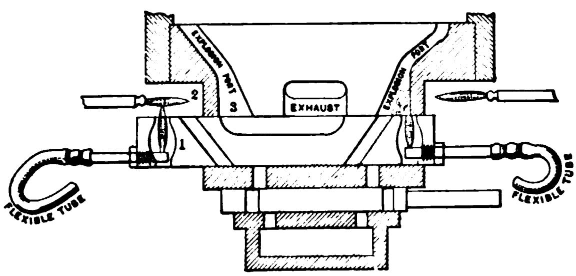


FIG. 11–11. — Hugon's Flame Ignition Valve.

in both fuel and lubricating oil consumption, but it did not find the degree of application of the latter, because apparently the means were not at hand to exploit it to the extent done with the Lenoir machine.

BEAU DE ROCHAS, 1862. — In a patent taken out in Paris by Alph. Beau de Rochas on January 7, 1862, he states the conditions required for the highest efficiency in an internal combustion

engine, and he also distinctly describes the working cycle which in his opinion is necessary to produce the highest efficiency.

According to this investigator, the conditions necessary for highest efficiency are four in number: (1) the greatest volume of the cylinder possible having a minimum surface of periphery; (2) highest possible velocity of motion; (3) greatest possible expansion; (4) greatest possible pressure at the commencement of the expansion. He states that for highest efficiency it is necessary to execute the following operations in the period of four consecutive strokes in each end of the cylinder.

(1) Aspiration during an entire out stroke of the piston.

(2) Compression during the following in stroke.

(3) Ignition at the dead point and expansion during the third stroke.

(4) Discharge of the burned gases from the cylinder during the fourth and last stroke.

The operations which are described above characterize the four-stroke cycle engine which was first actually built by Otto in 1876 or 1877. The importance of the pamphlet of Beau de Rochas was not recognized and it was probably little read until Otto had established the practical value of this method of operation. Five years before his death in 1887 Beau de Rochas was given a prize by the Société de Encouragement pour l'Industrie Nationale as a recognition of the important part which he had played in the development of the internal combustion motor.

Although the importance of compression previous to ignition in increasing both the efficiency and capacity of the internal combustion engine was fully pointed out by Million in his English and American patents and by the pamphlets of Beau de Rochas, little or no practical progress was made in the construction of compression engines for the next twelve years. The practical applications were confined in a large measure to the production of non-compression engines of the same type as that of Lenoir.

Otto and Langen. — Early in 1861, N. A. Otto, in trying to improve Lenoir's engine by giving it a full power stroke, hit upon the idea, in the course of his investigations, to break the igniter circuit of one of these machines, keep the exhaust valve closed at the end of the out stroke, then to turn the engine back in the other direction by hand, and when the piston reached the

inner dead center, having compressed the charge, to cause the spark to jump. He thus half unconsciously reproduced the suction and compression strokes of our modern four-cycle engines, and the result was that the engine received a sudden impulse which kept it spinning for several revolutions. As near as Otto thus came to the solution of the question, as proposed by Beau de Rochas in 1862, practical difficulties encountered, together with his own failures to realize the full importance of the thing, caused him to turn aside and invent an engine of an entirely different type. Not until thirteen or fourteen years later did he return to the old problem with success.

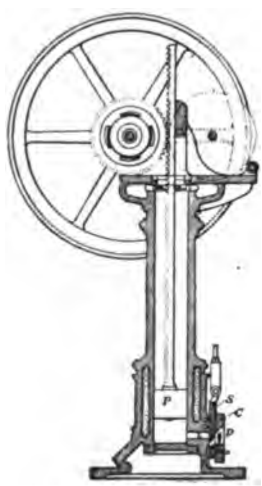


FIG. 11-12.—Otto-Langen Free Piston Engine.

The result of his labor in the early sixties was the so-called free piston engine. The idea had been already worked out by Brown in 1832 and by the Italian Barsanti in 1858, but it remained for Otto and Langen to make it a commercial success.

After several years of experimentation the first machine was exhibited at the Paris Exposition in 1867.

Flame ignition was used and much better economy was obtained than with the Lenoir or Hugon engine. Clerk states that "it completely crushed Lenoir and Hugon and held almost sole command of the market for ten years, several thousands being constructed in that period."

A section of the engine is shown in Fig. 11-12. It consists of a tall vertical cylinder water-jacketed through a part or the whole of its length with the top open to the air. The fly-wheel shaft is supported by bearings on the top and carries a gear which engages with a rack attached to the piston rod of the engine. The gear turns freely on the shaft while the piston is moving upward, but is connected with it by an ingenious clutch when the piston moves downward. A slide valve S, operated by an eccentric on a shaft geared to the main shaft, controls the admission, ignition, and exhaust intermittently as determined by the governor of the engine. When operating at full load the piston is lifted a few

inches and takes in the charge through the slide valve, which soon moves further and brings in the igniting flame. The resulting explosion projects the piston upward with high velocity. The pressure beneath it rapidly falls by expansion until lower than that of the atmosphere. On the return stroke of the piston the clutch is in engagement with the shaft and performs work due to the weight of the piston falling freely into a partial vacuum. The exhaust gases in the meantime are displaced through a port in the valve.

The clutch of the Otto-Langen engine was a very ingenious construction and one of the main points of the invention. It is shown in detail in Fig. 11–13 and consists of a part *a* keyed to

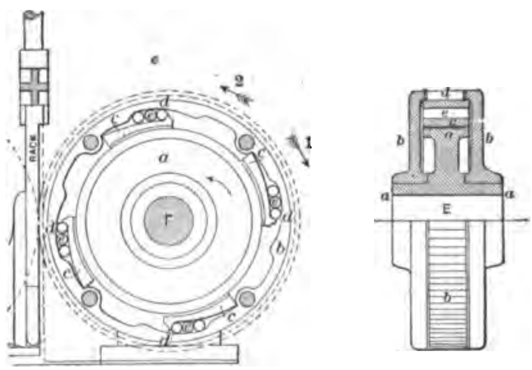


FIG. 11–13. — Clutches, Otto-Langen Engine.

the shaft, on which runs the part *b* carrying the teeth engaging the rack. The part *a* revolves freely with the shaft and is disconnected from the part *b* while the piston is moving upward or is stationary, but is fastened to the part *b* when the piston is moving downward. The parts *a* and *b* are engaged by small rollers, *e*, moving over wedge-shaped clips, *cc*, when the part *b* moves in the same direction as the part *a* and at a higher rate of speed. The parts are clamped together by the rollers wedging in between the two inclined surfaces and the steel clips *cc*. It will be seen that as soon as the part *a* moves with a higher velocity than is imparted to *b* by the rack, the clutch releases automatically.

Dr. R. H. Thurston, in the report of the Vienna exhibition in

1873, states that several of these engines were on exhibition and the results of several series of tests made by M. Tresca are given. The dimensions of the engine tested were as follows: diameter of piston, 8.75″; maximum stroke, 41.3″; diameter of water jacket, 15.75″; height of water jacket, 28.0″. The following are the results of four trials:

Trials	1	2	3	4
Pressure of gas, millimeters	30.5	36	36	....
Duration of experiments, hrs.	4	1	0.5	0.5
Revolutions per min.	85.7	82	81.5	79.9
Brake H. P.	0.896	0.857	0.426	0.418
Total gas per hr., litres	1017.5	1085	560	560
Gas per H. P. hr., litres	1135.6	1266	1314	1339.7
Gas per H. P. hr., cu. ft.	39.4	44	45.5	46
Gas used for igniters per hour, cu. ft.	2.12	2.30		

The above results, however, have been surpassed in tests by Meidinger in 1868. The engine tested had a cyl.-diameter of 5.9″, a maximum stroke of 38.7″, and was rated at ½ horse-power. At maximum load this engine developed .635 B. H. P. and showed a gas consumption, including that of the ignition flame, of 29.5 cu. ft. per B. H. P. hour. This corresponds to a thermal efficiency on the brake of 13.7 per cent. Curiously enough a still better economy was shown at a lower load. With a B. H. P. of .35, the efficiency was 15.4 per cent. The fuel mixture in both cases contains 12.5 per cent of gas.

Respecting the engine tested at Vienna, Dr. Thurston states that it has always worked well and has greatly reduced the consumption of gas over that used by earlier engines. It is very noisy in operation and the exceptional fuel economy is its only special recommendation. This economy is probably due principally to the extreme rapidity with which the piston is projected upward at each explosion of gas, the work of expansion being thus utilized before sufficient time has elapsed for any serious amount of condensation to occur by contact with cold surfaces. A diagram from a 2 horse-power Otto-Langen engine from Clerk's work on the gas engine is shown in Fig. 11–14. This diagram shows that the expansion during the working stroke continues below that of the atmosphere, the piston evidently being taken to the

top of its stroke by the energy stored in the fly-wheel at the instant of explosion.

The igniting device used consisted of a constantly burning flame outside the working cylinder which was put in communication by a slide valve with the explosive charge at the proper time. The large sizes of this engine were governed by controlling the velocity of discharge of the exhaust gases. The more the exhaust outlet was throttled, the slower the descent of the piston, and hence the fewer the number of cycles in unit time.

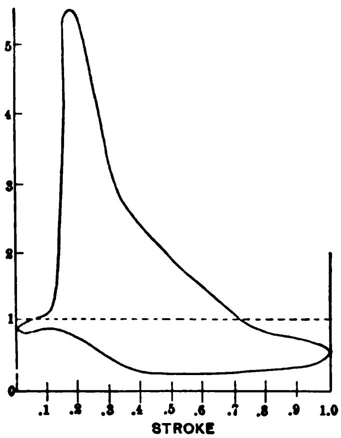


FIG. 11-14. — Diagram, Otto-Langen Engine.

BRAYTON. — In point of time the next important invention in the art of producing internal combustion motors was made by an American, George B. Brayton. His engines were built in large numbers and possessed many advantages over any previously made. They had the remarkable distinction of being the only motors which had been designed in which ignition takes place at constant pressure, the form of the diagram, already shown in Chapter I, being similar in many respects to that of the steam engine.

Brayton took two American patents for his engines; first the earlier one dated April 2, 1872, covered an engine adapted to burn gas; the later one, patented June, 1874, covered an engine adapted to burn liquid hydrocarbon or petroleum. Both engines have essentially the same principle of operation, the air and combustible are supplied to the working cylinder under pressure for a portion of the stroke, ignition continues during the admission of the air and combustible, causing an increase of volume without change of pressure. The supply valve closes when sufficient combustible has entered, and the piston finishes its stroke by expansion. Ignition was produced by a constantly burning flame in the power cylinder. Wire diaphragms were used to keep the flames from striking back.

Of the two forms of Brayton engine, the gas engine never

found much application on account of the fact that, due to puncturing the diaphragm, the ignition flame would sometimes strike back and explode the compound mixture stored in the reservoir.

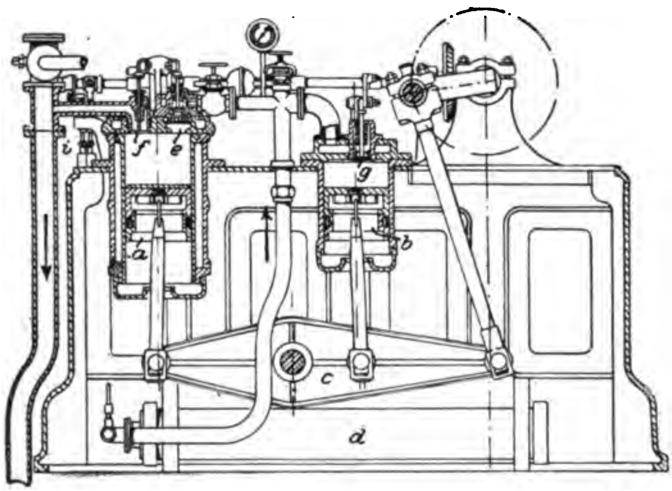


FIG. 11–15.— Elevation, Brayton Oil Engine.

This action together with the high price of gas led to the development of the petroleum engine. Figs. 11–15 and 11–16 show the construction of this engine very clearly.* The engine

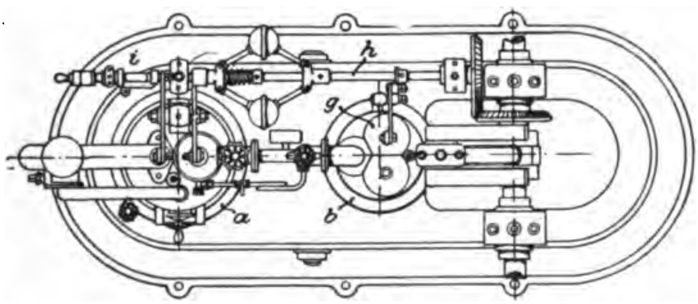


FIG. 11–16. — Plan, Brayton Oil Engine.

consists of a power cylinder *a* and an air-compressor cylinder *b*, the pistons of which are connected to the walking beam *c*. The air, drawn into the cylinder *b* through an automatic valve, is

* Güldner, Entwerfen und Berechnen der Verbrennungsmotoren, p. 91.

discharged through the valve *g* into the receivor *d* when the pressure has reached the desired point. From *d* the compressed air flows into the power cylinder when the valve *e* is opened by the lever shown, through the action of a cam on the auxiliary shaft *h*, Fig. 11–16. On its way through the valve the air is saturated with the vapor of gasoline or petroleum by passing a layer of saturated felt held between two perforated plates. The small pump *i*, shown in both figures and actuated from the cam shaft *h*, serves to keep the felt saturated. Just below this carbureting device, a needle flame, kept supplied with gasoline by the pump *i*, is kept constantly burning, igniting the charge as it passes. The admission valve is opened as the working piston reaches its upper end of the stroke. The mixture therefore burns as the piston is traveling downward, and the rate of combustion is so regulated that the pressure remains practically constant during the time of admission Fig. 11–17 shows a typical work diagram. The card from the air cylinder is of course just like the ordinary air-compressor diagram.

FIG. 11–17. — Power Card, Brayton Engine.

The engine was governed by cutting off the mixture at the desired point. This was accomplished by making the admission valve cam conical and sliding it along the shaft *h* by means of the governor shown, thus varying the time of opening of the valve.

This oil engine was a thoroughly practical machine and found considerable application. Regarding its economy, a test by Clerk showed the following figures:

Diameter of Motor Cylinder	8″
Stroke of Motor Cylinder	12″
Diameter of Air Cylinder	8″
Stroke of Air Cylinder	6″
Mean R. p. m.	201
Mean B. H. P.	4.26
Net Indicated H. P.	5.40
Pump H. P.	4.10
Total I. H. P.	9.50
Mechanical Efficiency, %	79.0
Petroleum pr B. H. P.-hour, gals	.323

The fuel consumption shown amounts to a thermal efficiency of 6 per cent on the brake.

THE OTTO ENGINE. — The Otto engine was patented by Nicholas A. Otto of Deutz, Germany, in the United States, August 14, 1877, and in England about one year earlier. The engine was shown with great success at the Exposition of 1878. It was called the "Silent" engine, probably in contradistinction to the free piston engine of the same inventor.

The patent taken out by Otto is devoted principally to a description of an improved method of introducing air and gas into the cylinder of a gas engine, in layers or strata so as to make the explosion less violent in its nature. The patent describes three kinds of engines to which his process could be applied, viz., a non-compression engine, an engine of similar structure to the non-compression engine but in which the charge is compressed by external means before passing into the cylinder, and lastly the now common four-cycle engine in which the compression is performed in the working cylinder and which followed in its mode of operation the principles laid down in the French patent of Beau de Rochas in 1862.

The Otto method of introducing air and gas in separate layers to produce slow combustion did not prove to be of practical value, but the construction described in Claim 3 of the American patent not only revolutionized all previous methods of gas engine construction but formed the basis for all subsequent practice. This claim is stated as follows:

> 3. A gas-motor engine wherein, by one out stroke of the piston, separate charges of combustible gaseous mixture and air are drawn into the cylinder, which charges are compressed by the in stroke and then ignited, so as to propel the piston, which by its return stroke expels the products of combustion.

The Otto engine was so greatly superior to all the earlier ones in economy and capacity for a given weight and regularity of operation that it soon distanced all competitors. Its success led to numerous infringements in Germany and England and in each of these countries a patent suit was brought; in Germany the Otto patent was declared invalid because of the earlier patent to Beau de Rochas, but in England it was sustained. In the United States no patent suit was brought, although there were doubtless many infringements before the patent expired in 1894.

Figures 11-18 to 11-20 show a horizontal cross-section, a side

view and an end view of an Otto engine of 1884.* In this engine the admission of the fuel mixture and its ignition are controlled by the slide valve, *b*. This valve is actuated by a crank, *d*, and moves forward and backward across the cylinder head. The gas

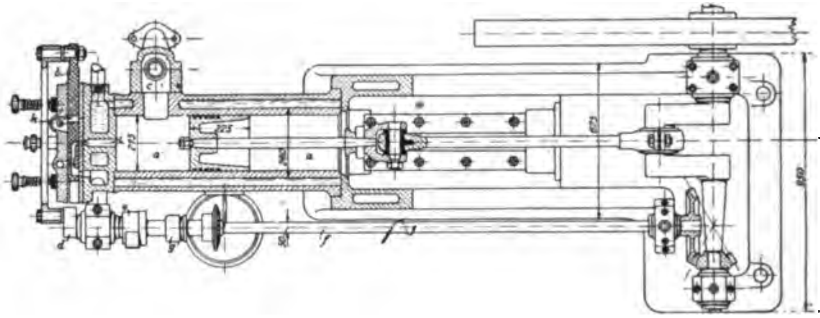


FIG. 11–18. — Horizontal Section Otto Engine of 1884.

valve, *g*, is actuated by a cam, *g′*, upon the side shaft, *f*, while the exhaust valve, *c*, of the now usual poppet type, is actuated through a lever by means of a cam, *e*, upon the same shaft. This shaft, *f*, turns with one half the speed of the crank shaft. The

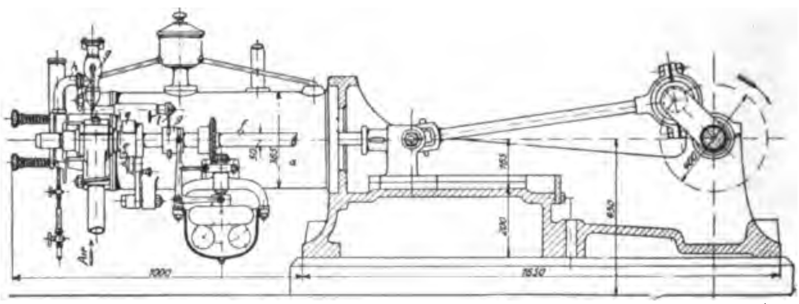


FIG. 11–19. — Elevation of Otto Engine of 1884.

admission slide valve is held against its seat by a cover plate. This plate contains the ignition arrangements and air and gas ports, which latter coincide with other ports through the valve proper and into the combustion chamber, *a′*, in certain positions of the valve. The gas valve is connected to the gas port in the plate by a bent pipe, *d*, while gas is furnished to the ignition apparatus by the small forked pipe shown in Fig. 11–20.

* From Güldner, p. 43.

To explain the operation of the engine, suppose that in the position shown in Fig. 11–18 the piston is just commencing its suction stroke. The exhaust valve, *c*, has just closed and the port in the valve coincides with the port, *i*, into the cylinder. As the piston moves outward, only air is drawn in for the first part of the stroke, because the cam has not yet opened the gas valve, *g*. A little later *g* opens and the mixture is drawn in for the rest of the stroke. At the outer dead center the valve has closed the port, *i*, and compression next takes place. By the time the piston has reached the inner dead center, the valve has moved far enough over to bring the ignition cavity, *k*, in front of the port, *i*. The flame in *k* strikes in, causing the charge to explode. Expansion and exhaust follow.

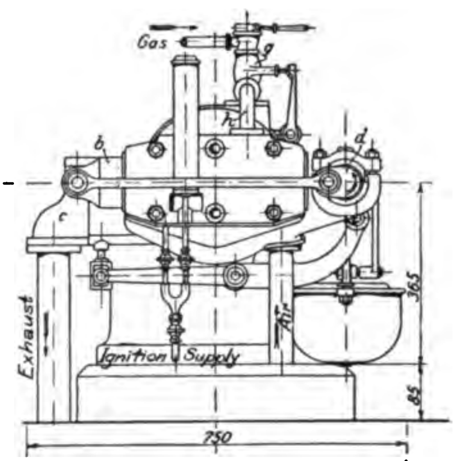


FIG. 11–20. — End View of Otto Engine of 1884.

The above method of drawing in the charge was by Otto supposed to result in stratification, *i.e.*, according to his views there would be a layer of burned gases next to the piston, then a layer of air and last the layer of mixture. He also claimed that this arrangement was not disturbed during compression. His argument was that, ignition taking place in the rich mixture, the pressure wave would soon reach the leaner mixture and the layers of air and burned gas in succession, and "tone down," so to speak, in its intensity so as to avoid shock to the engine mechanism. The opinions of experts regarding the soundness of this theory are divided even to-day. See Chapter IV.

The ignition apparatus used in the earlier Otto machines, up to the general introduction of electric ignition, was very similar to that employed by Otto in the free-piston engine some ten years earlier. Fig. 11-21 shows a cross-section through the ignition

cavities in plate and slide. One branch of the forked pipe, shown in Fig. 11–20, supplies a constantly burning flame, *G*, Fig. 11–21, while the second branch fills the cavity, *B*, with gas which gets its air supply through the port, *C*, the mixture igniting when it strikes the flame, *G*, as shown. When near the time of ignition, cavity *B* in the slide is cut off from its air and gas supply, but enough burning mixture is left in the cavity to ignite the charge in the cylinder when *B* registers with the port into the cylinder. Since the pressure in the compression chamber is so much higher than that in the cavity, there is danger that the flame will be blown out when communication is first established. To prevent this, just before the cavity and the inlet port commence to register, communication is established with the combustion chamber through a very fine opening, thus equalizing the pressures in inlet port and cavity.

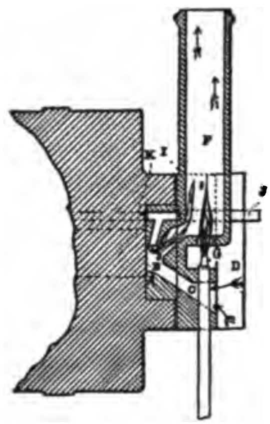


Fig. 11–21.—Ignition Arrangements, Otto Engine of 1884.

The governing arrangements were simple and effective. The gas valve cam, *g′*, Fig. 11–19, was arranged to slide on the shaft, *f*. A fly-ball governor controlled its position, and when the speed rose a certain amount above normal, the cam was pulled far enough to the left to cause it to miss the valve lever, *l*. Thus the engine received no gas and an impulse was missed.

Economy tests on early Otto engines were made by Slaby and Brauer, 1881, Teichman and Boecking, 1887, and by others. The thermal efficiencies on the brake ranged from 9 to 12 per cent. The best figure obtained by Brauer in 1886 on a Deutz 4 horsepower horizontal engine was a total gas consumption of 29.9 cu. ft. per B. H. P. hour, which corresponded to a thermal efficiency of 15.5 per cent. Tests made by Brooks and Stewart at Stevens Institute, Hoboken, in 1882 on a 6 B. H. P. engine show about the same results.

The engine shown in Figs. 11–18 to 11–20 is from the design standpoint a well-built machine, and shows a very marked advance in this respect over all earlier forms. It will be noted that

the cross-head was still used. This was, however, soon dispensed with, substituting a trunk piston, and thus shortening the length of the machine. Another change soon instituted was the substitution of electric for flame ignition.

CLERK. — The disadvantage, inherent in all four-cycle machines, of receiving a power impulse only once in four strokes, led other inventors to experiment with the two-cycle engine which gets a power impulse every turn. The change from the four- to the two-cycle principle seems very simple, but is in reality

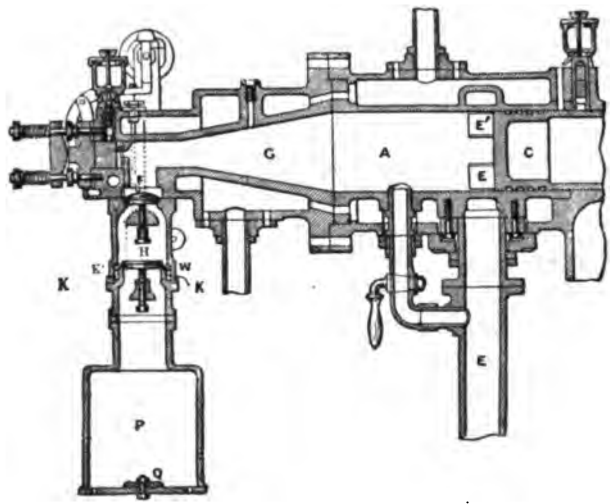


FIG. 11-22. — Vertical Section, Clerk Engine.

beset with many difficulties. Perhaps the earliest fairly successful worker was Clerk, who commenced to experiment upon two-cycle engines soon after Otto perfected his four-cycle machine. It was, however, not until 1880 that he succeeded in producing a serviceable machine. Figs. 11-22 and 11-23, both from Clerk, "The Gas and Oil Engine," show a vertical and horizontal cross-section respectively. The operation of the engine is as follows: In Fig. 11-23 the power piston, C, has just reached the outer dead center and the main bulk of the exhaust gases have escaped through the ports $E\,E'$. In the meantime the displacer piston, D, which on its previous suction stroke has drawn a mixture of air and gas into B, has completed about half of its in stroke and

displaced the mixture into G and A through the connecting pipe, W. About the time D has completed its in stroke, the power piston, C, has covered the exhaust ports on its return stroke, and compression ensues in the main cylinder. The cylinder volumes are so proportioned that in theory no mixture can be lost through the exhaust ports. At the inner dead center of the power piston, C, or just before, the igniting cavity, O, Fig. 11–23, comes opposite the port, N, and the charge is fired. The piston is impelled forward and the next charge, drawn into B in the meantime, commences to enter the space G as soon as the pressure in A has fallen enough, after the beginning of exhaust, to cause the valve in the pipe between B and G to lift. The intermediate valve

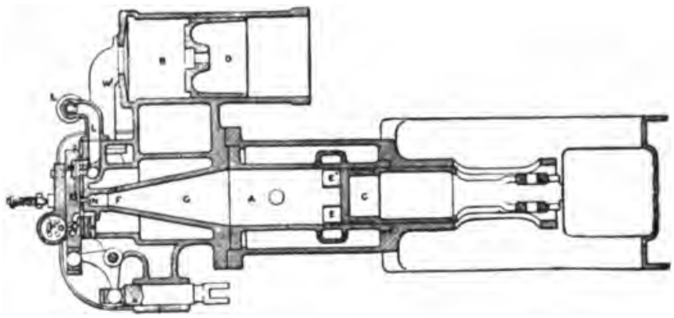


FIG. 11–23. — Horizontal Section, Clerk Engine.

arrangement is shown in Fig. 11–22. P is an air chamber. The air drawn in by the suction of the displacer piston, D, passes through the valve, H, and in so doing is mixed with gas which enters through a number of fine holes in the seat of the valve from the annular space, K. On the in stroke of the displacer piston the mixture under some pressure comes in under the valve, F, which it lifts as soon as the pressure of the exhaust gases above it has fallen low enough.

The difficulty under which the engine labored was loss of mixture through the exhaust ports and consequent low economy. Although the respective volumes of the two cylinders may be in the proper ratio, the charge expands by heat on the transfer, and mechanical agitation favors the loss. In spite of this defect shop tests on 2, 4, 6, 8, and 12 horse-power engines in 1885 gave results fully equal to those obtained on the four-cycle machine

of that day. The following table shows the results of some of these tests.

Horse-power	2	4	6	8	12
Dia. Motor Cyl., inches	5	6	7	8	9
Stroke Motor Cyl., inches	8	10	12	16	20
Dia., Displacer, inches	6	7	7½	10	10
Stroke Displacer, inches	9	11	12	13	20
R. P. M.	212	190	146	142	132
I. H. P.	3.62	8.68	9.05	17.38	27.46
D. H. P.	2.70	5.63	7.23	13.69	23.21
Mech. Eff., %	74.7	65.0	80.0	78.8	84.5
Gas per B. H. P.-hr., cu. ft.	40.0	37.3	30.42	26.58	24.12

Figures 11–24 and 11–25 show a power diagram and a pump diagram from a 6 horse-power engine.

4. **Period of Application.** — It is intended in what follows to give merely a brief résumé of the further development of the internal combustion engine say up to 1897 and to reserve a more detailed description of the most important forms in the market to-day for the next chapter.

FIG. 11–24. — Power Card, Clerk Engine.

On account of the Otto patent, which practically amounted to a monopoly, other manufacturers were forced to turn their attention to the development of the two-cycle engine. How Clerk solved the problem with fair success has been already shown. He was followed by Wittig and Hess (1880), Benz (1884), Söhnlein, Güldner (1893–1898), Oechelhäuser (1896) and Koerting (1898) in Germany, Robson, Southal, and Samson in England, Benier (1894) in France, and Mietz and Weiss in the United States. Many of these engines are in the market to-day, and some are described in the next chapter.

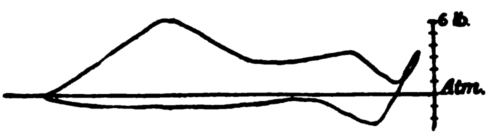


FIG. 11–25. — Pump Card, Clerk Engine.

With the fall of Otto's claims in Germany about 1885 the field

became free, and many manufacturers of the two-cycle engine abandoned it for the more simple four-cycle. The fall of this patent was, in a sense, a misfortune as far as the two-cycle engine was concerned, as it held back the development of that type of machine at least ten years. It is only within the last six or eight years that the very obvious advantages of the two-cycle principle again received the attention they deserve. On the other hand, the development of the four-cycle engine after 1885 was extremely rapid. Thus while the limit of power was about 4 B. H. P. in 1878 and units of from 15-20 horse-power could be had in 1880, the limit soon rose to 100 horse-power in 1889 and 200 in 1893. Blast furnace gas called for units of 600 horse-power in 1898, while to-day engines developing up to 4000 horse-power are being built. With the increase in size up to the neighborhood of 200 horse-power, there comes an increase in thermal efficiency. Thus a Crossley engine of 12 horse-power soon showed a gas consumption of 24.3 cu. ft. of illuminating gas per B. H. P. hour. To-day efficiencies exceeding 25 per cent on the brake, with lean power gases, are not rare, and Güldner, by intelligently applying sound principles of construction, has succeeded in obtaining economic efficiencies exceeding 30 per cent.

It would lead too far for the scope of this book to describe even a fair percentage of the various four-cycle engines brought out between 1885 and 1898. The most prominent names connected with the development of these engines are perhaps Loutzki in Germany (1888), Delamare-Debouteville & Malandin, who in 1900 brought out the first large blast furnace gas engine, in Belgium, Charon in France, Crossley Brothers in England (1892 and 1898), and Westinghouse in the United States (1896). Four-cycle constant-pressure oil engines were developed by Capitaine (1889-91), Brünnler (1893-4), and Diesel (1893-97) in Germany. The Diesel engine is to-day one of our most important internal combustion engines, and is considered in greater detail below. Among the delevopers of the four-cycle constant volume oil engine, Daimler's work in connection with the high-speed engine deserves special mention. Other engines of this type brought out in this period are Spiel (1884), Capitaine (1885-90), Priestman (1889), Hornsby-Akroyd (1892), Banki (1894), and Haselwander (1898).

One of the greatest achievements of this period is the development of the Diesel engine.

DIESEL. — The history of the Diesel engine is interesting. It began in 1893 when Rudolf Diesel, in a pamphlet entitled "Theory and Construction of a Rational Heat Motor to replace the Steam Engine and other existing Heat Engines," laid down the following "fundamental requirements for a perfect combustion."

1. Attainment of the highest temperature in the cycle, not by means of combustion and during the same, but before and independent of it by compression of air alone.

2. Gradual injection of atomized fuel into this highly compressed and heated air so that during combustion no rise of temperature takes place, *i.e.*, the combustion shall be isothermal. For this purpose the process of combustion cannot after ignition be left to itself, but must be governed from the outside to maintain proper relation between pressure, volume, and temperature.

3. Correct choice of weight of air with reference to the heating value of the fuel and the desired compression temperature, so that the practical operation of the machine, lubrication, etc., shall be possible without water-cooling.

It is interesting to follow out these points and to see in how far their object has been attained.

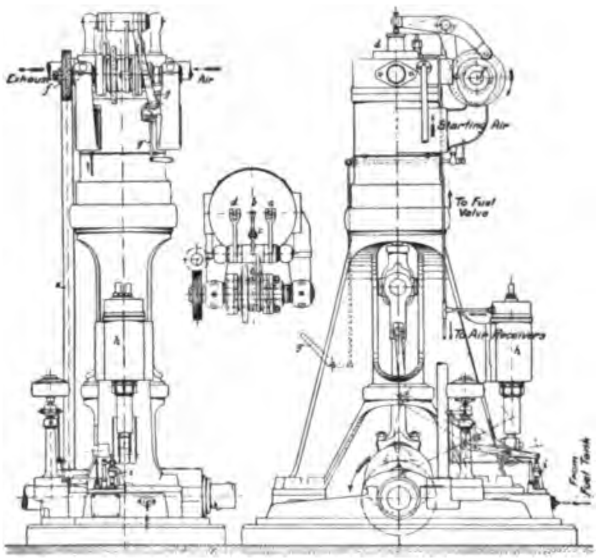
The intended fuel was coal dust, the cycle the Carnot. At the very outset, however, a modification was made in the cycle in cutting out the isothermal compression and substituting for it one stage adiabatic compression. But a jacket was not thought necessary, and in fact a non-conducting lining for the cylinder was demanded.

As a consequence of the above pamphlet, two firms, Krupp in Essen and the Maschinen-fabrik Augsburg, undertook the construction of experimental machines. As was to be expected, further changes from the original idea were necessary, the two most important of which were the substitution of oil for coal dust, and the use of a water jacket.

In 1898 the experimental stage had been so far passed that Schröter could report test figures which more than doubled the thermal efficiency of the then existing Otto engines. The final form of the engine as now constructed is shown in Figs. 11-26 to

11–28, which represents a Diesel engine built by Krupp in 1898.*

The internal-cylinder construction offers nothing new; *a* is the suction valve for air, *b* the fuel valve, and *d* the exhaust valve. *c* is the starting valve not in commission during ordinary operations. All are actuated through levers by cams on the shaft, *f*, which is operated from the crank shaft through the intermediate shaft, *e*; *h* is the air pump to furnish compressed air for fuel injection and for starting. Both of these are taken from steel



FIGS. 11–26 to 11–28. — Diesel Engine, 1898.

flasks into which *h* delivers; *i* is the oil pump under control of the governor which regulates the amount of oil per stroke to the load.

On the first down stroke, the engine takes air through *a* and compresses it on the return stroke to a pressure of about 460 pounds, with a temperature of about 1100 degrees Fahrenheit. Just before the end of the up stroke the fuel valve, *b*, is opened to a width of only a few hundredths of an inch, and the injection air, previously compressed to about 650 pounds by pump *h*, flows

* Güldner, p. 101.

into the compression chamber, carrying with it and finely atomizing the oil furnished by the pump, *i*. The oil ignites on entering, due to the high compression temperature. In spite of the fact that isothermal combustion is intended, the lack of outside control causes a rise not only in the pressure of from 80–100 pounds, usually not very noticeable on the indicator card, but also a rise of temperature approximating 1800 degrees Fahrenheit. Thus while the intended mean temperature of the cycle of Diesel's pamphlet was about 350 degrees, that realized in actual operation is about 950 degrees Fahrenheit. The time during which the fuel injection valve, *b*, remains open, is constant for all loads, but the effective stroke of the fuel pump, *i*, ends the sooner the lower the load, thus effecting regulation.

After the closing of *b*, expansion commences and is followed by exhaust through *d* on the return stroke.

The engine is started by the compressed air furnished by *h* during a previous operation and stored in a tank. To start, the cams on the shaft, *f*, are pulled to the right by the lever, *g*. This puts the valves *a*, *b* and *d* out of commission and starting valve *c* in commission. On placing the crank just beyond the upper center, and opening the tank valve, the engine takes compressed air for a few turns just like a steam engine takes steam. When the required momentum has been obtained, the cams are released and are snapped back into place by a spring at the proper time.

The history of the development of the Diesel engine is interesting in that the final construction departs so far from the patented ideal that the engine of to-day does not seem to be protected by the claims of that patent. The fact that no igniter was necessary was only incidentally considered by Diesel, and is not expressly covered by the patent.

The first test figures on a Diesel engine were published by Schröter in 1897. The dimensions of the engine were: Cylinder diameter, 9.85″, stroke 15.7″, rated B. H. P., 18–20. The fuel used was American kerosene having a heating value of 18400 B.T.U. The following table shows the results of two full load trials and compares them with some of the theoretical results figured by Diesel in his pamphlet of 1893.*

* Güldner, p. 107.

	Diesel Engine of 1897		Ideal Engine of 1893
Load	Full	Full	....
R. P. M.	171.8	154.2	300
B. H. P.	19.87	17.82	?
Total I. H. P.	27.85	24.77	100
Pump H. P.	1.29	1.17	....
Net I. H. P.	26.56	23.69	100
Oil per B. H. P.-hr., lbs.	.543	.523	.247 lbs. of coal per I. H. P.
Cooling water pr B. H. P.-hr., lbs.	154	203	Zero
Compression Press., Atm.	32.5	32.5	250
Max. Comb. Pressure, Atm.	36	36	250
Temp. End of Compression, °C.	550	550	800
Temp. End of Combustion	1600	1600	800
Indicated Thermal Eff., %	33.7	34.7	73
Mech. Eff., %	71.0	72	....
Thermal Eff. on Brake, %	25.2	26.2	?

It is clear from the above table that, although the engine did not realize the early expectation of its designer, the results shown are a remarkable step in advance as regards the economy of the then existing internal combustion engines. To-day even better figures are frequently obtained. E. Meyer, for instance, has lately tested a Diesel engine showing an indicated thermal efficiency exceeding 42 per cent. Fig. 11-29* shows the general form of the indicator card obtained from Diesel engine. This is a regulating diagram, the size of the card decreasing with the load by shortening the cut-off.

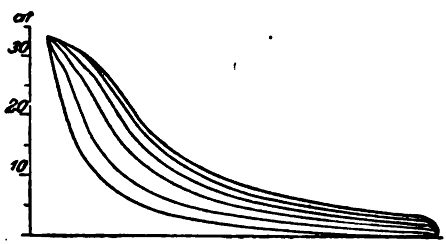


FIG. 11-29.—Regulation Diagram, Diesel Engine.

* Güldner, p. 109.

CHAPTER XII

MODERN TYPES OF INTERNAL COMBUSTION ENGINES

THE previous chapter brought the development of the internal combustion engine up to 1897. Since then expansion has been very rapid, until to-day we find that the design of gas engines has been standardized in the most important particulars just as was the case with the steam engine some decades ago. It is intended in the present chapter to give a brief description of the most important engines found in the market to-day. An examination of the market will show some fairly definite divisions among manufacturers as far as size of engine made is concerned. Thus there are but a half dozen firms in this country, and a few more than this in Europe, who make engines up to the very largest sizes. It is comparatively easy, therefore, to describe nearly all of these various engines, and this is very desirable on account of their importance. Next we find a somewhat larger number of medium sized engines of various types, and lastly a very large number of engines up to say 50 horse-power serving the general commercial field for small powers. It is of course impossible to cover the last two classes of engines to any great extent. The majority of these engines do not differ except in minor details, and for these reasons the following list has been confined to what seem to be the most representative machines of each class.

Regarding the *general features of design*, small engines are either horizontal or vertical. The cylinders are almost invariably single-acting, multiplication of power being obtained by increasing the number of cylinders. The vertical offers some advantages over the horizontal form, in that the foundation need not be as large or as heavy. Further it is claimed that it is easier to lubricate the cylinders uniformly, and that the wear on the cylinder is less. A favorite form of frame for this type of engine in this country is the box frame with enclosed crank case, using splash lubrication. Some European designers object to this

form, claiming that all supervision of crank pins and intermediate bearings is by this form of frame rendered impossible. The small two-cycle machine almost invariably uses the enclosed crank case for the pre-compression of the mixture.

What has been said of the small machine applies in general also to medium sized engines. Vertical machincs here possess the added advantage that it is easier to dismount them by means of overhead crane than is thc case with horizontal machines. The limit to a vertical engine comes in the head room required. For this reason all of the very large machines, as well as medium sized double-acting machines which require a cross-head, are horizontal. Another reason that may be cited is that it is easier to operate a medium sized or large horizontal engine than it is a vertical because all climbing or mounting platforms is avoided, and the whole installation is more completely under the operator's eye. Finally, the use of some of the industrial power gases favors the use of the horizontal machine, because any dust carried can be much more easily swept out of a horizontal than a vertical cylinder during regular operation.

The double-acting engine is perhaps not used as widely as it deserves to be, increase in power being generally sought by multiplying the single-acting cylinders. There is, however, to-day no reason why double-acting cylinders are not as reliable as the single-acting. For large machines, double-acting cylinders are almost an economic necessity, and the clumsy four-cylinder double-opposed large engine has become thoroughly obsolete. The largest engines of to-day are double or twin two-cylinder tandem double-acting engines.

The very obvious disadvantages of the trunk piston can be quite successfully overcome for small and medium sized machines, and hence it is almost universally employed for these sizes. But for large machines the use of the trunk piston is indefensible, being opposed alike by considerations of manufacture and of reliable operation. The fitting of large pistons of this type offers grave difficulties in the shop, and in operation proper lubrication is difficult; further, the main office of the piston is to confine the gases without leakage; to make it also act as the machine member to take up the lateral thrust of the connecting rod may, in large machines, seriously interfere with its main purpose.

The crank shaft of small and medium sized engines is nearly always of the center-crank type. This type is very rigid and, above all, transmits the stresses equally to both sides of the frame. In double or twin machines, however, such a shaft would call for four main bearings, the proper alignment of which might cause some trouble in large engines of this type. For this reason, some American makers prefer the side-crank shaft, which is a much less costly shaft to make, and reduces the number of bearings for a twin engine to two. Of course, the side-crank frame takes up the explosion stresses eccentrically, and therefore has to be designed heavier than the center-crank frame. On account of this fact and the generally higher stresses in gas engine frames as compared with steam-engine frames, European designers will not use the so-called side crank or Tangye form of frame.

The period of experimentation in design, and of freak design, has largely passed in gas-engine practice, and, as mentioned at the outset of this chapter, standardization has made welcome progress during the last few years. The alcohol engine, now in its development as far as the United States is concerned, does not call for any radical changes in the existing designs of liquid fuel engines. As far as further progress is concerned, it is not unlikely that the next step will be the development of a highly efficient constant-pressure gas engine after the manner of the Diesel liquid fuel engines, taking up the thread again where Brayton left it in the seventies.

A. Gas Engines

1. SMALL AND MEDIUM SIZED ENGINES. — Among the best known makers of small and medium sized gas engines in this country may be mentioned the Otto Gas Engine Works of Philadelphia, makers of the Otto engines; the Fairbanks-Morse Company; the Jacobson Machine Manufacturing Company, of Warren, Pa.; the Jacobson Engine Company, of Chester, Pa., the Struthers-Wells Company of Warren, Pa., makers of the Warren engine; the Bruce-Meriam Abbott Company, of Cleveland; the Westinghouse Machine Company; the Olds Gas Power Company, of Lansing, Mich.; the De LaVergne Machine Company of New York, makers of the four-cycle Koerting engines; the Weber Gas Engine Company of Kansas City, Mo.; the A. H. Alberger Company of

Buffalo, makers of the Buffalo tandem engines, the Foos Gas Engine Company of Springfield, Ohio, etc.

In most cases the small gas engine operates on illuminating gas, natural gas, or gasoline, but in many instances attachments are furnished so that the engine can be run on either gas or liquid fuel as desired. Producer gas is not usually employed for very

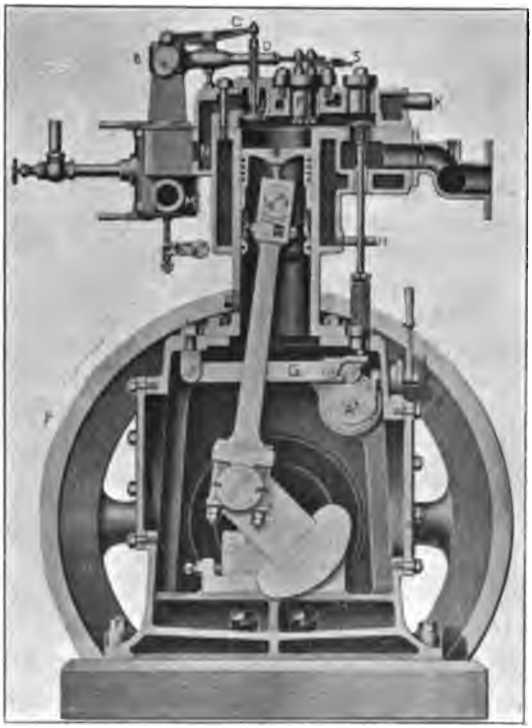


Fig. 12-1. — Westinghouse Engine.

small units. A 10 to 15 horse-power suction gas producer for hard coal is the present lower limit and perhaps the exception, while soft coal producers in their present state of development do not run less than 50 to 60 horse-power.

The Westinghouse Gas Engine. — The cross-sectional cut, Fig. 12-1, shows the essential parts of the Westinghouse vertical engine. In this type the crank mechanism is completely enclosed, and splash lubrication is depended upon for the proper oiling of

the intermediate bearings, the crank pins and the pistons. Both the inlet and exhaust valves are mechanically operated by cams and shafts driven from the main shaft; the inlet valve, *J*, by means of cam *B* and lever *C*, the exhaust valve *E* by cam *A* and the roller lever shown. The igniter, operated from the inlet cam shaft, is located at *F*. Gas and air, after passing the chamber *M* in the proper proportion, enter the passage *N* on their way to the inlet valves. The engine is governed by means of a governor of the fly-ball type which regulates the amount of mixture entering the passage *N* (see Chapter XIV). The smaller sizes of this

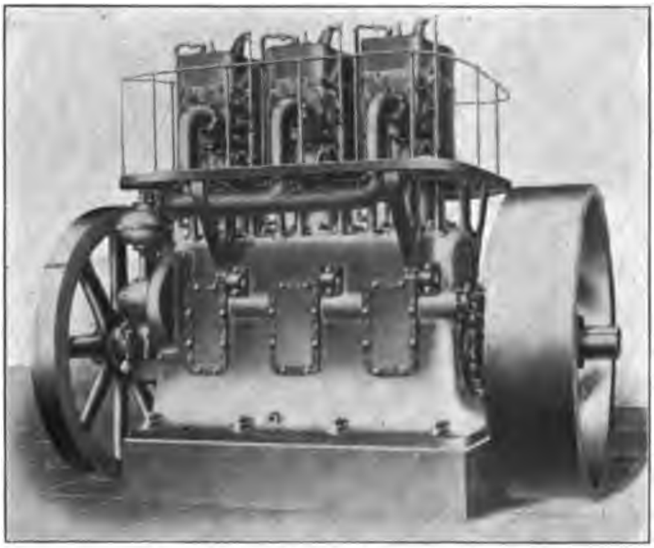


FIG. 12-2. — Westinghouse Engine.

machine have two cylinders, and can generally be started by hand; the larger sizes, above about 85 horse-power, have three cylinders and are generally started by compressed air, which is admitted to one cylinder, starting the engine, while the other cylinders operate normally. Figs. 12-2 and 12-3 show general views of three-cylinder machines.

Engines manufactured under the Jacobson Patents. — There are two firms manufacturing engines under these patents, the Jacobson Engine Company of Chester, Pa., and the Jacobson

Machine Manufacturing Company of Warren, Pa. The engines made are of three types, hit-and-miss, automatic cut-off, and throttling. The Warren Company make hit-and-miss engines from 2½ to 25 horse-power and automatic engines from 8 to 27 horse-power. The Chester Company make hit-and-miss engines from 30 horse-power up, and automatic cut-off and throttling engines from 33 horse-power up. All of these engines so far mentioned are single-acting. The automatic cut-off and throttling

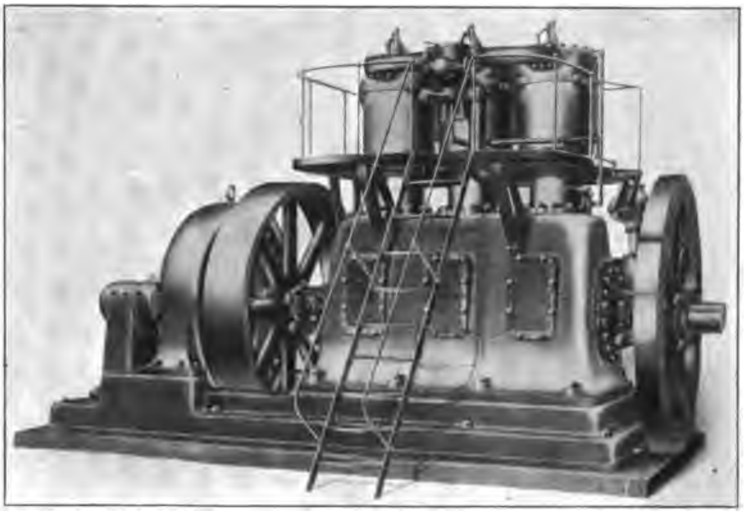


FIG. 12-3. — Westinghouse Engine.

types are built as single-cylinder or as tandem or twin-tandem units. The Chester Company now also undertake the building of double acting automatic cut-off or throttling engine as tandem or twin-tandem units up to any power desired.

The structural features of the engines built by the two companies mentioned are of course very nearly the same, so that one description will do for both.

The general features of the hit-and-miss machine are shown in Fig. 12-4. The most interesting detail of the design is perhaps the removable cylinder bushing, which is an unusual but highly commendable construction for small engines. This not only allows of choosing the proper grade of metal for the cylinder barrel, but thermal stresses are also avoided by admitting of

free expansion, in this case against the packing at the head of the bushing. On account of the manner of holding it, this packing can never be blown out.

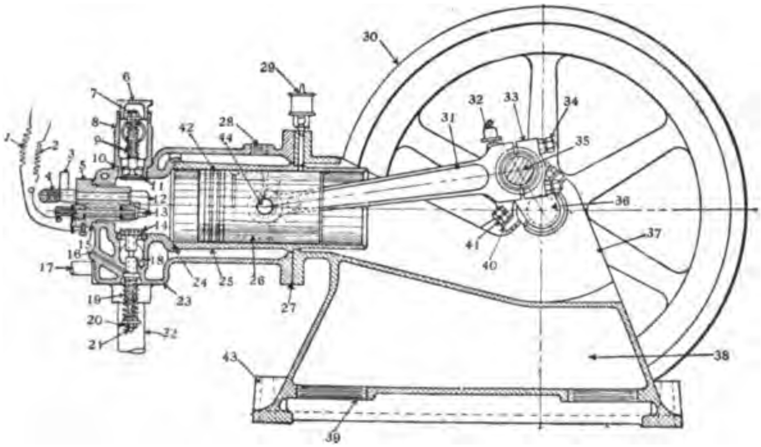


FIG. 12-4. — Jacobson Hit-and-Miss Engine.

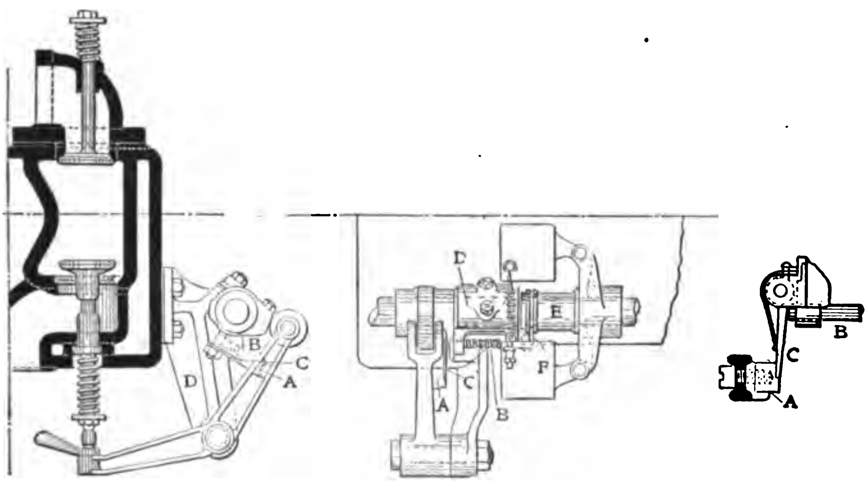


FIG. 12-5. — Governing Mechanism of Jacobson Hit-and-Miss Engine.

The inlet valve is automatic, and is held in a separate housing. The manner of operating the exhaust valve and the governor control are shown in Fig. 12-5. The side shaft is operated by means of screw gears, which is the most satisfactory way of

transmitting the motion. The governor is of the fly-ball type and is operated by the lay shaft. When the speed becomes too high, the blade, C, Fig. 12–5, put in position by the governor, engages the block A on the exhaust valve lever, prevents this lever from returning, thus holding the exhaust valve open.

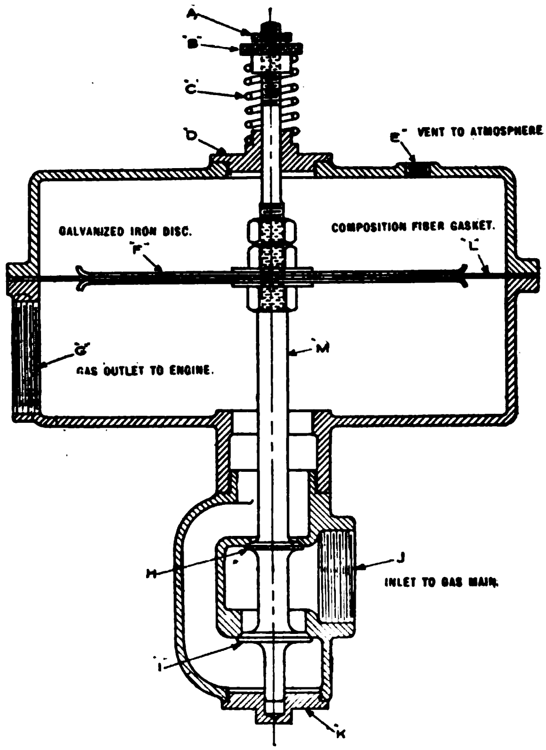


FIG. 12–6. — Gas Pressure Regulator for Jacobson Engines.

Figure 12–4 also shows the make-and-break igniter. One block contains both electrodes and the location of the spark is central as regards the volume of the charge.

This engine can be run on either gas or gasoline, depending upon whether a gas regulator or a carbureter is employed. The type of regulator used is illustrated in Fig. 12–6. Its construction is very simple, there is nothing apparently to get out of order. It is claimed that this regulator furnishes the gas to the engine

under atmospheric pressure at all times. The firm makes an attachment which allows of the change from one fuel to another at a moment's notice.

The general design of the automatic cut-off is the same as for the hit-and-miss engine, the difference being in the governor and the valve gear. Fig. 12–7 shows the method of operating the inlet and exhaust valves and the igniter. The valve gear is shown in greater detail in Fig. 12–8. The seats and stem bushings are all easily accessible and replaceable. The exhaust valve is operated in the ordinary way by a cam from the lay shaft. For the inlet valve lever an eccentric is used. As the eccentric rod *R* rises, it pushes upward the valve lever *Q* at the right end and opens the inlet valve. The inlet valve spindle carries two discs, the main inlet valve and the gas valve. Just before the exhaust valve closes, the gear commences to open the inlet valve, but only air enters since the gas valve *V* remains closed. The air thus entering serves to scavenge out the combustion chamber. A moment later the valve spindle has descended far enough for the set collar on the spindle to depress the gas valve, after which the mixture begins to enter the cylinder. Both valves close simultaneously when the nose of the eccentric rod *R* is forced off the end of the valve lever by the action of the inclined plane *P*. The position of this plane is determined by the governor, which is of the fly-ball type and directly driven from the main shaft. Thus the valve always opens at the same point, but it closes sooner or later, depending upon the load.

Fig. 12–7. — Jacobson Automatic Cut-off Engine.

Figure 12–9 shows the general appearance of the automatic cut-off machine. They are built in sizes up to 27 horse-power by the Warren Company, and can be adapted to run on gasoline and natural, illuminating, or producer gas. The twin tandem type is shown in Fig. 12–10 and is built in sizes above 100 B. H. P. for producer gas and above 120 B. H. P. for natural gas by the

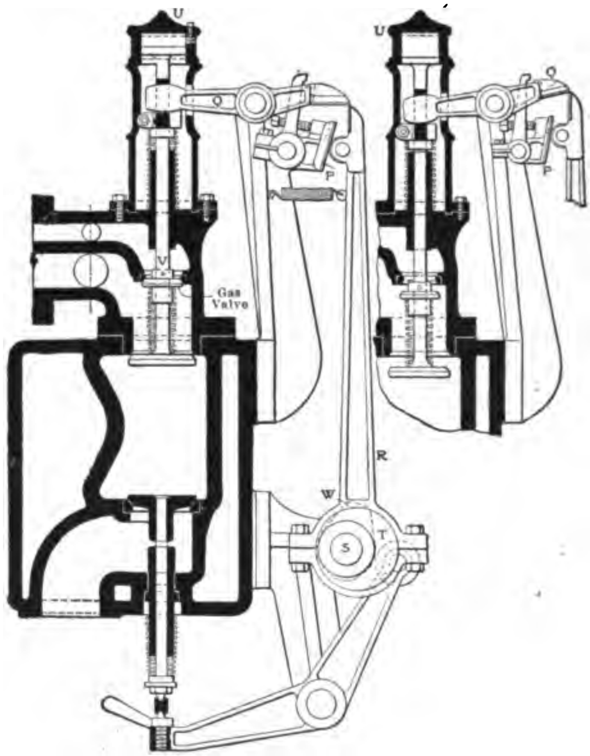


FIG. 12-8. — Valve Gear, Jacobson Automatic Cut-off Engine.

Chester Company. One-half of this unit, the tandem engine, is built in sizes from 60 B. H. P. upward for natural gas and from 50 B. H. P. upward for producer gas.

FIG. 12-9. — Jacobson Automatic Cut-off Engine.

The Chester Company's throttling engine, a general view of which is shown in Fig. 12-11, differs from the automatic cut-off machine only in some details of valve gear. A cross-section of this engine is shown in Fig. 12-12, while Fig. 12-13

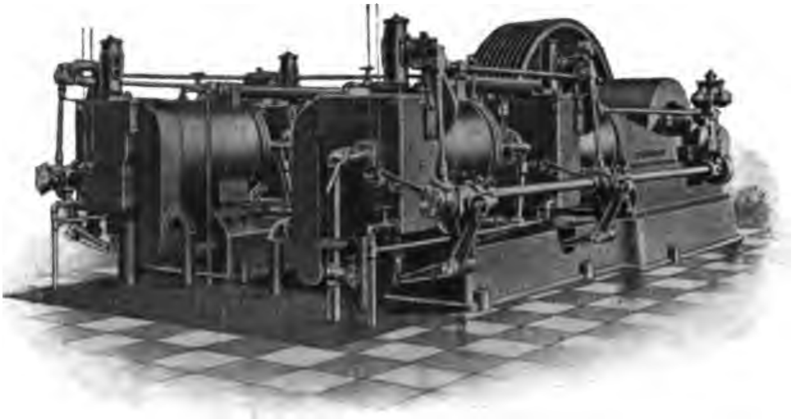


FIG. 12-10. — Jacobson Twin-tandem Automatic Cut-off Engine.

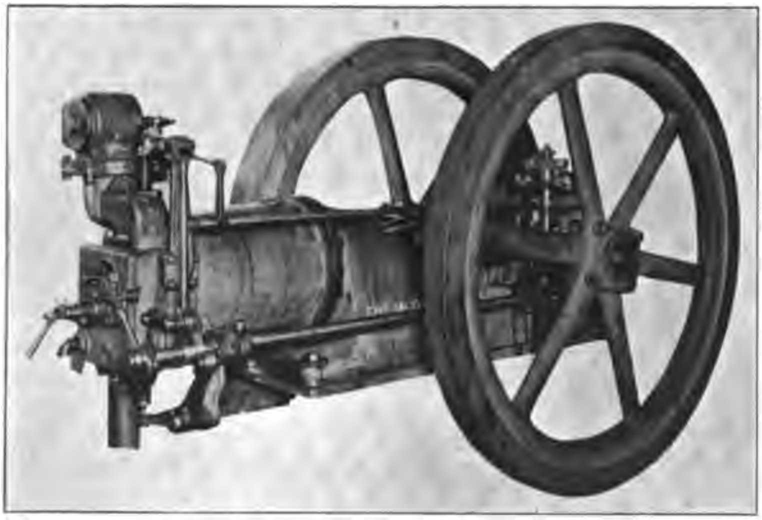


FIG. 12-11. — Jacobson Throttling Engine.

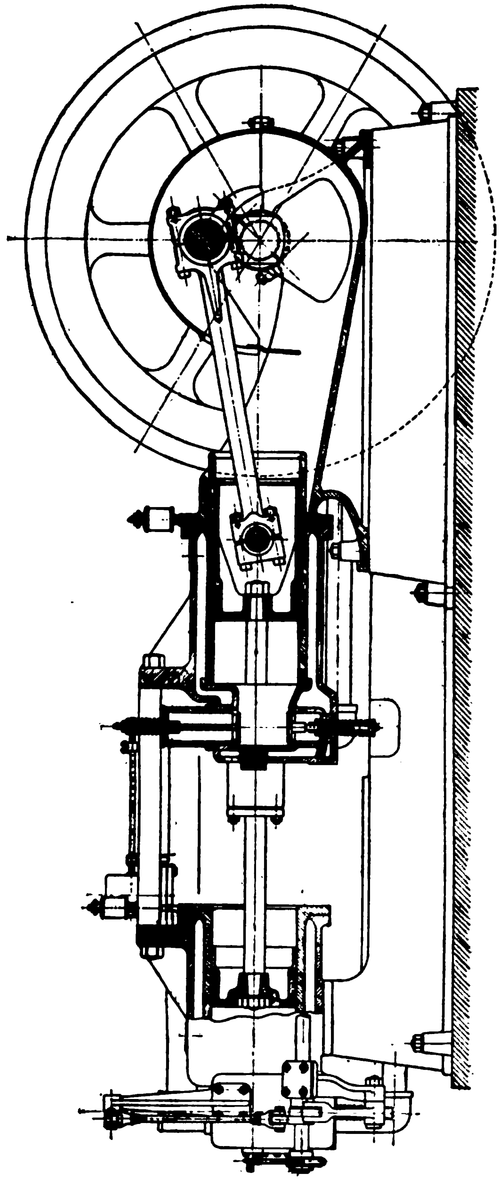


FIG. 12-12. — Jacobson Tandem Throttling Engine.

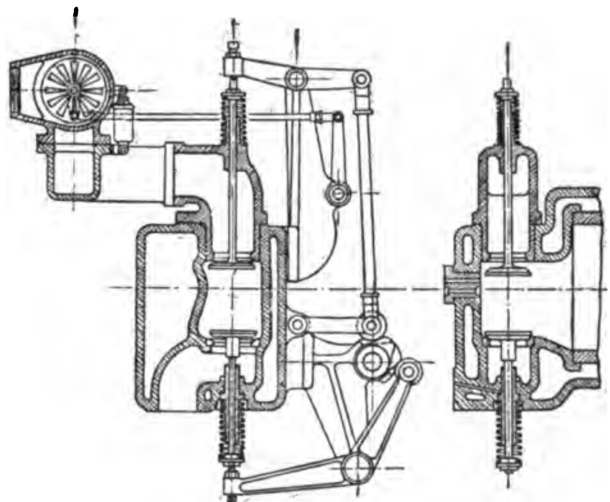


Fig. 12-13. — Valve Gear, Jacobson Throttling Engine.

illustrates the valve gear. Both valves are operated by cams. The charge passes the mixing and throttling valve, which is controlled by the governor through the reach rod shown.

Fig. 12-14. — Bruce-Meriam-Abbott Engine.

The Bruce-Meriam-Abbott Engine. — The Bruce-Meriam-Abbott Company, located in Cleveland, Ohio, manufactures engines for natural, illuminating, and producer gas and for gasoline. The design used for natural and illuminating gas and for gasoline is shown in Fig. 12-14. The sizes range from 12 to 250 horse-power, two-cylinder up to 125 horse-power, and four-cylinder above that size. The design of cylinder and frame is along conventional lines. The cam shaft across the top of and between

the cylinders is operated through spur and bevel gears as shown. The valves are of the poppet type and are located in the head. Above 55 horse-power they are held in separate cages which are easily removable. The cam shaft operates these valves by rocker arms on either side. This construction is more clearly shown in Fig. 12–15. An excellent feature of the machine is the purely cylindrical form of the combustion chamber.

FIG. 12–15. — Detail of Bruce-Meriam-Abbott Engine.

Governing is effected by a governor of the fly-ball type operated by the lay shaft. The governor sleeve operates one end of a lever passing between the cylinders, the other end supports the mixing valve, the details of which are shown in Fig. 12–16. Gas and air enter the annular space shown. On the suction stroke of the engine the gas flows into the space surrounding the piston valve and mixes with the air by flowing out through the port about half way up. The two combined then enter the interior of the valve through the six ports shown; from this chamber the mixture goes to the engine. The position of the valve controls the amount of throttling and thus regulates the weight of the charge going to the engine. It is stated that from full to no load this valve has to move only $\frac{1}{16}$ inch.

FIG. 12–16. — Mixing Valve, Bruce-Meriam-Abbott Engine.

Jump spark ignition is used. The method of supporting the spark coil and the system of wiring is well shown in Fig. 12–15. The timer is very simple. There are two copper pins each about $\frac{1}{4}$-inch diameter and $\frac{3}{4}$ inch long. One of them projects from the bottom of a small cup while the other is fastened to the end of a flat spring, and dips down in the same cup. The spring may be seen at *A*, Fig. 12–15. The cup is filled with oil and the points are normally held apart a small distance. Just before a spark is desired the spring is

depressed by means of a cam, the circuit is made by bringing the points in contact and the spark is produced at the moment the cam releases the spring. What amounts to an outside spark gap (see next chapter) is provided so that the spark may be watched.

To convert any gas engine of this design into a gasoline engine it is necessary merely to furnish a fuel pump and to replace the iron piston valve of the mixer by brass or bronze to prevent rusting. To retain the high compression used with gas, however, this company has adopted the Bánki principle of injecting water into the cylinder. The device is said to give entire satisfaction.

For producer and natural gas this firm makes engines ranging from 25 to 200 horse-power, two-cylinder units up to 90 horse-power, and four-cylinder above that. The design of these engines is apparently somewhat different from those above described, Fig. 12–17, the principal change being that the lay shaft evidently runs alongside the cylinders instead of between them. The cylinder and frame construction is otherwise the same, but the interesting feature about the design is the fact that the center line of the cylinder is offset about one-half the length of the crank from the center line of the main bearing, as near as can be scaled from the drawing. This is on the principle of the Ramsey crank mechanism, the idea of which is to equalize the wear due to the side thrust of the piston against the cylinder on both sides of the cylinder and to improve the turning moment. Both of these aims are attained with a moderate offset such as here used.

The Fairbanks, Morse & Company Engines. — The Fairbanks, Morse & Company manufacture a number of different types of engines, both vertical and horizontal, for gas and liquid fuel. The general features of the horizontal design are shown in Fig. 12–18. It appears from this that the exhaust valve, at the side of the cylinder, is mechanically operated, while the inlet valve placed in the head is automatic. The governor is placed in the fly-wheel and may be either of the throttling or hit-and-miss type. In the latter case it operates to hold the exhaust valve open. The ignition gear is of the make-and-break type, arranged so that at starting the spark may be retarded.

Engines above 9 or 10 horse-power are fitted with a self-starting device which consists of two parts, a match detonator and a hand pump. Both are shown in plan in Fig. 12–18 at the side

of the cylinder. The hand pump serves to pump a combustible mixture into the cylinder, with the crank just beyond the center, which mixture is fired by igniting a parlor match inserted into the detonator. The pressure so generated is sufficient to start the engine under some load.

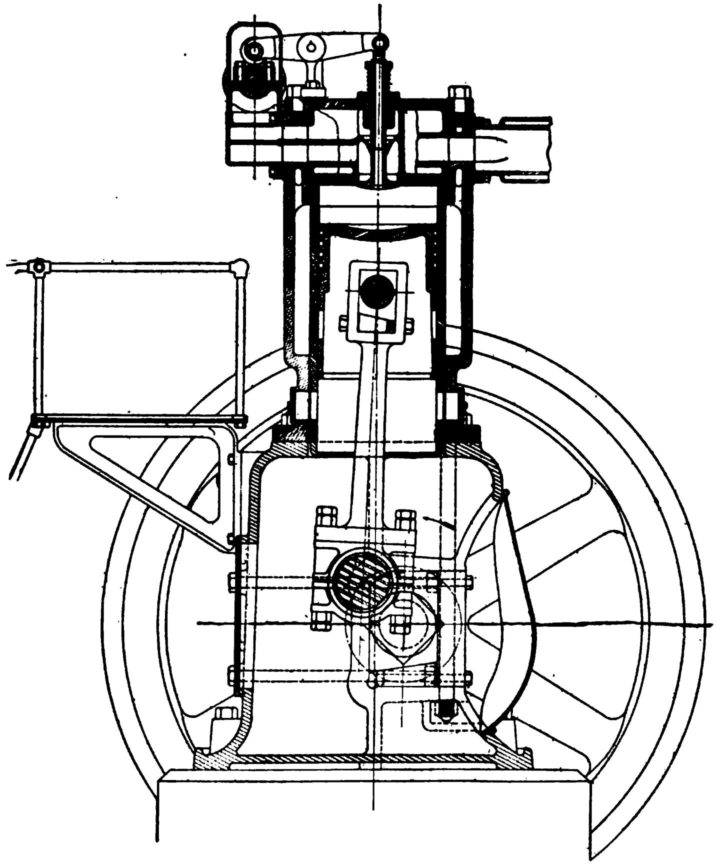


FIG. 12-17. — Bruce-Meriam-Abbott Engine for Producer Gas.

Engines are built for gasoline, naphtha and distillate, for kerosene, for alcohol, and for gas. In each case, of course, the fuel feeding and mixing arrangements differ somewhat. In the liquid fuel engines the feeding device acts positively, a pump being used to inject the proper quantity of fuel into the charge of air. In

the gas engine there is instead a mechanically operated gas valve. If desired the engine may be arranged to run on either gas or

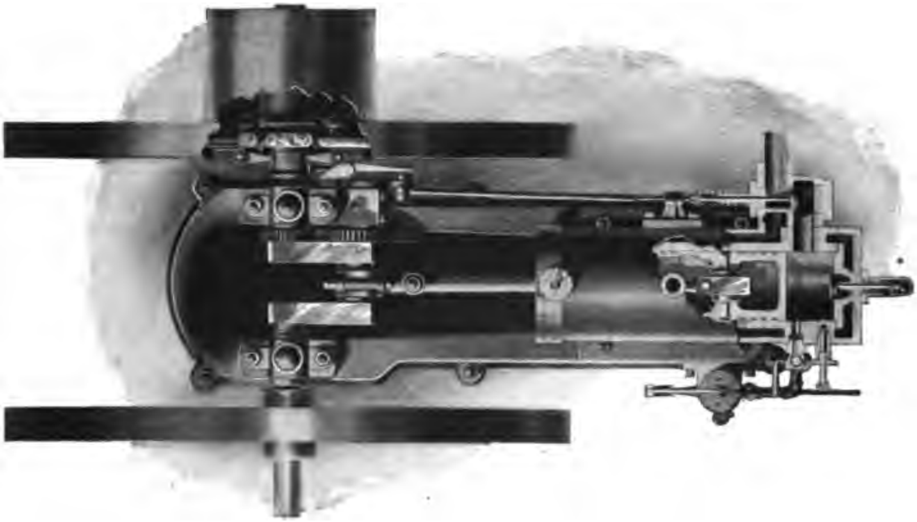


FIG. 12-18. — Fairbanks-Morse Engine.

liquid fuel. Fig. 12-19 illustrates an engine having these features.

The same company also builds two distinct types of producer gas engines. The first of these has the general features of the standard horizontal engine above described, the second is quite different, as shown in Fig. 12-20. In this design both valves are of the vertical poppet type, mechanically operated from the side shaft, the inlet valve on top, the exhaust at the bottom. As in the other Fairbanks engines, the ignition is by make and break. The governor is of the fly-ball type, operated from the lay shaft. Regulation is by throttling a mixture of constant proportion.

FIG. 12-19. — Fairbanks-Morse Engine for Liquid or Gas Fuel.

The general type of Fairbanks vertical engine is shown in Fig. 12-21. These engines are built for any kind of fuel.

Governing is effected by throttling the mixture; make-and-break ignition is used. Although the crank case is enclosed, lubrication is positive instead of by the splash method usually employed in this design.

FIG. 12-20. — Fairbanks-Morse Producer Gas Engine.

The Koerting Four-cycle Gas Engine. — This German machine is manufactured in this country by the De La-Vergne Machine Company of New York. The general features of the design are clearly shown in elevation, Fig. 12–22, and the two cross-sections, Figs. 12–23 and 12–24. The entire design gives the impression of being very substantial and thorough. The frame is a very rigid construction and the cylinder is supported by the frame throughout the length. The cylinder head is a somewhat complicated casting. The inlet and outlet valves are placed vertically over each other and are operated by cams from a lay shaft. The combustion chamber is divided by a water-cooled tongue projecting from the cylinder head. The purpose of this projection is to effectually cool the interior of the combustion chamber and to thus draw down the compression temperature, admitting of higher compression. The mixing valve, shown at the left of the transverse section in Fig. 12–24, is automatic. Regulation is effected by means of a governor of the Hartung type which

FIG. 12–21. — Fairbanks-Morse Vertical Engine.

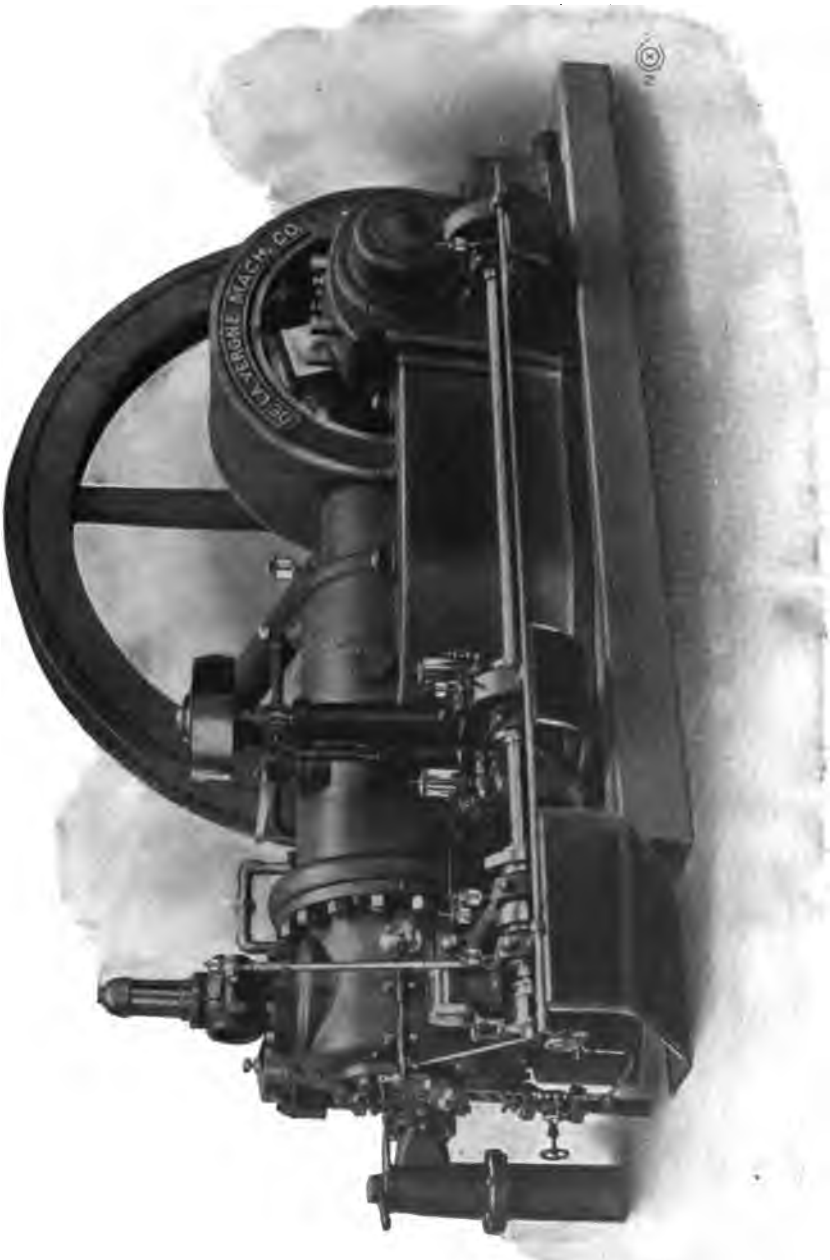


FIG. 12-22. — Koerting Four-cycle Gas Engine.

operates a butterfly throttle valve in the admission passage, as shown. The speed is thus controlled by throttling. In engines exceeding 100 horse-power the pistons are water-cooled. All engines above 12 horse-power have electric igniters while those

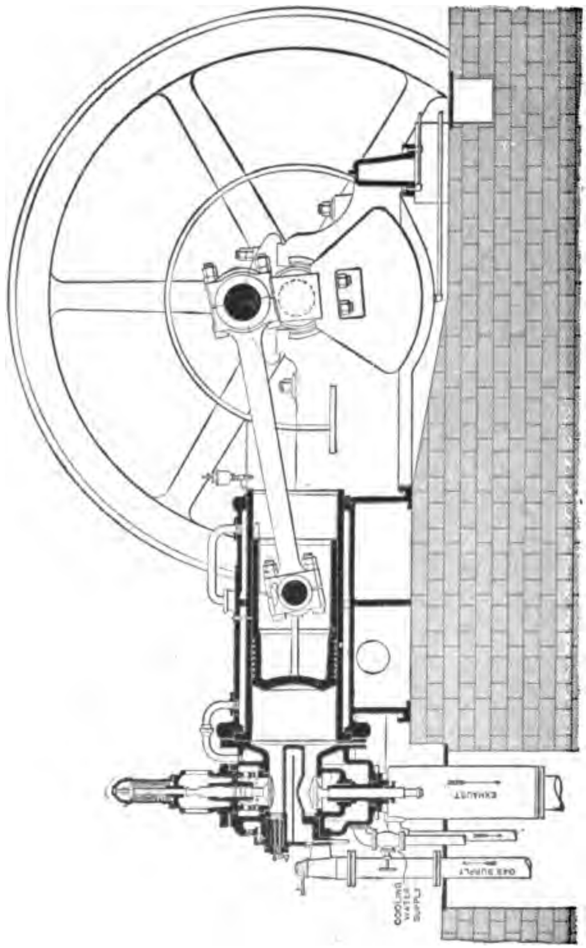


Fig. 12-23. — Longitudinal Section of Koerting Four-cycle Gas Engine, showing Cylinder Construction and Valves.

below this power use the hot tube, at least as constructed by the parent firm.

The Buffalo Tandem Engine. — This engine is made by the A. H. Alberger Company of Buffalo. A full description will be

found in an article in Power for February, 1907, from which the following illustrations are taken. Fig. 12–25 shows a general view, Fig. 12–26 a vertical cross-section, and Fig. 12–27 two cross-sections of the valve chest at right angles to each other.

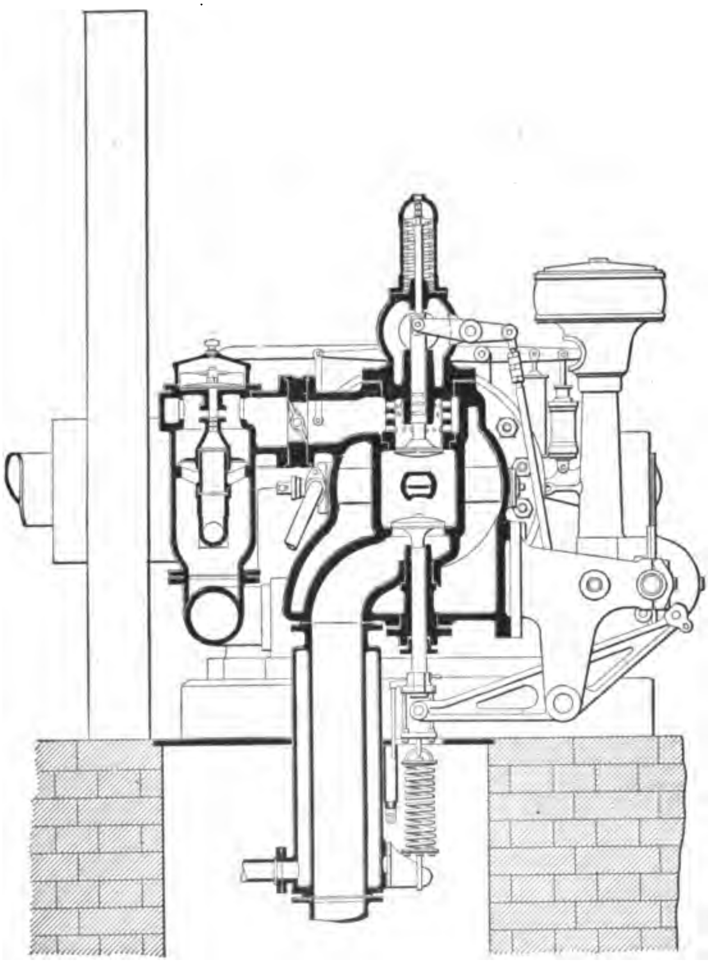


FIG. 12–24. — Valve Gear, Koerting Four-cycle Engine.

The engine differs materially from those previously described in having two cylinders in tandem, each acting on the four-cycle principle. The back piston is made like an ordinary steam engine

piston since there is no side thrust in this cylinder. Its piston rod passes through a water-cooled stuffing-box as shown. The back cylinder head is a simple flat plate not water-cooled; all

FIG. 12-25. — Buffalo Tandem Engine.

parts requiring it, however, are thoroughly cooled. The valves are of the double-guide poppet type and are placed side by side in a valve chest at the side of the cylinder, Fig. 12-27. Cams

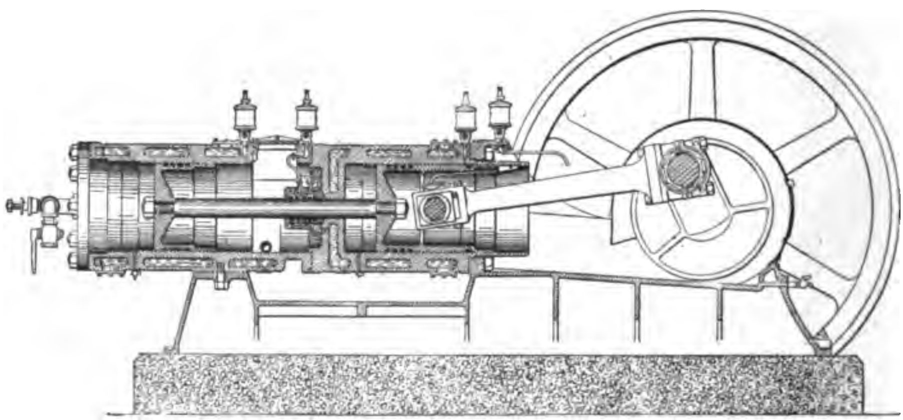


FIG. 12-26. — Section Through Cylinders and Pistons of Buffalo Gas Engine.

on a lay shaft operate these valves, as shown in the transverse section, Fig. 12-27. The make-and-break igniter, which is adjustable during operation, is placed over the inlet valve at the

right side of the valve chest. The exhaust pipe is fastened at the left side as shown in the longitudinal section of the valve

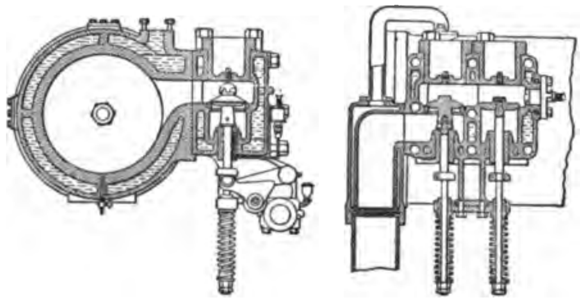


FIG. 12-27. — Valve Details of Buffalo Engine.

chest. The jacket water, after passing through the jacket, is made to enter the exhaust pipe, cooling the gases and thus acting as a muffler. The mixing and governing arrangements of this

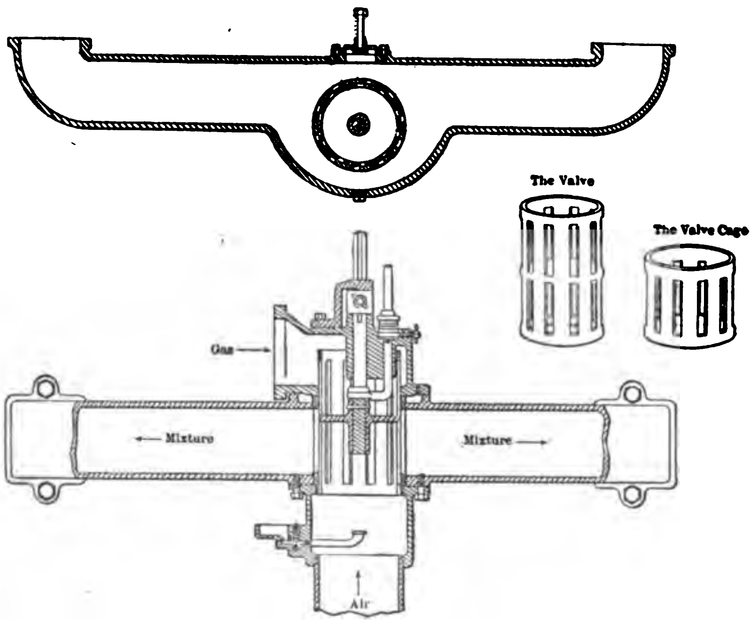


FIG. 12-28. — Mixing and Governing Arrangements Buffalo Engine.

engine are shown in Fig. 12-28. The two inlet valve chambers are connected by a header, as shown in Fig. 12-25. At the center

this header carries the mixing valve. This valve, Fig. 12–28, is a hollow cylinder divided into two parts by a transverse partition. Each half has a number of slotted ports which in certain positions of the valve register with other similar ports in the cage. The valve has two offices. Its position up or down in the valve cage controls the ratio of air to gas. If moved up the effective gas port area is reduced, while that of the air ports is increased by the same amount, and vice versa. Thus no matter what the gas used, the total effective area is in all cases the same. Rotary motion of the valve controls the cut-off of the mixture along the suction

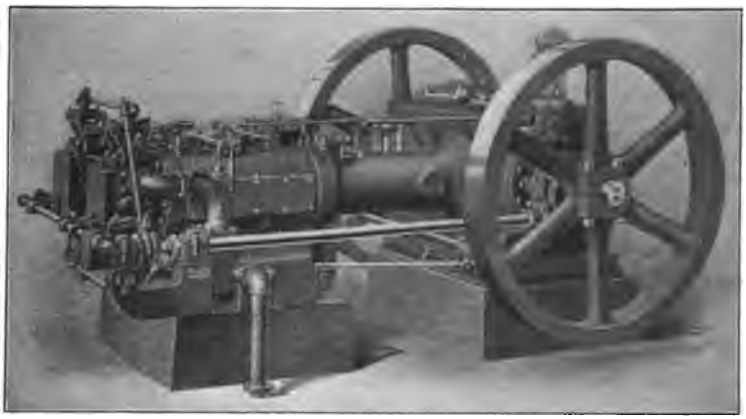


Fig. 12–29. — Buckeye Two-cycle Engine.

stroke. This action is controlled by a Rites inertia governor which operates through linkage as shown in Fig. 12–25.

The Buckeye Two-cycle Gas Engine. — This engine, made by the Buckeye Engine Company of Salem, Ohio, illustrates a type of medium sized two-cycle engine. The following description is taken from Power, September, 1906. It is a single-acting scavenging twin engine and can be made to operate also on gasoline or distillate.

The present engine has two motor cylinders, with cranks set at 180 degrees, two fuel pumps and two air pumps. Fig. 12–29 shows the side of the engine on which the secondary shaft is located and gives a good general view of the valve gear. Fig. 12–30 is a sectional elevation of one element of the twin engine. The piston 2 performs a double duty; in addition to delivering

the power of the explosions to the crank, it compresses the charge in the chamber 18 for delivery to the combustion chamber. The cross-head 6 is in the form of a plunger and acts as an air pump and compressor piston in the chamber 7. The exhaust ports are opened by the piston 2, as usual in engines working on the two-stroke cycle. These ports are shown at 14, 14, Fig. 12–30. Each piston compresses its own explosive mixture, but each cross-head plunger delivers compressed air for scavenging to the combustion chamber of the other half of the engine unit. The cycle of operation is as follows:

When the piston uncovers the exhaust ports, the scavenging valve 11 is opened by the valve gear and compressed air at about 8 pounds per square inch is admitted to the combustion chamber from the air pump of the other cylinder. This air blast sweeps out the

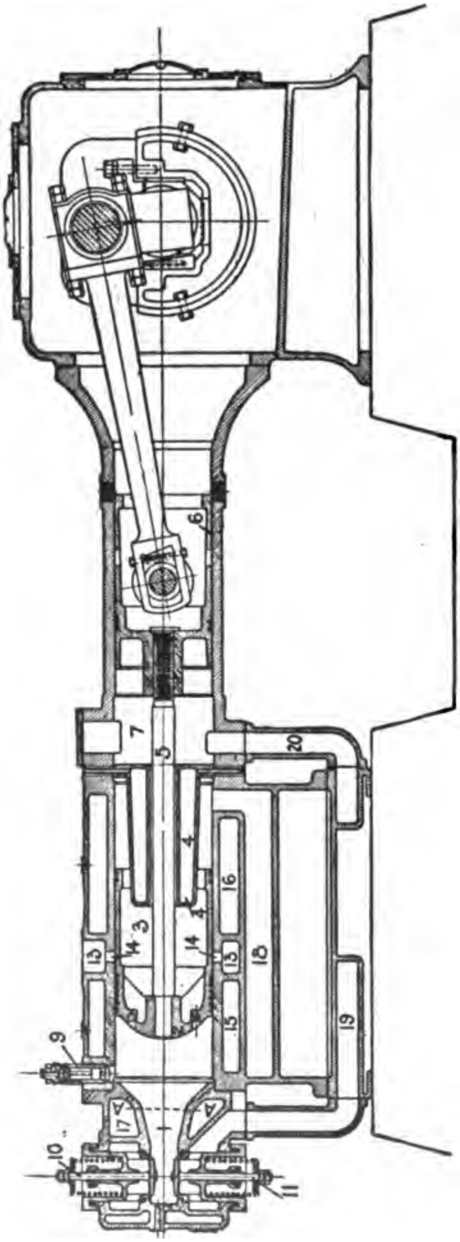


FIG. 12–30. — Section of Buckeye Two-cycle Engine.

burnt gases, and the admission valve 10 is then opened, admitting a charge of gas and air from the compression chamber 18; this charge is also at a pressure of about 8 pounds per square inch. The piston further compresses the charge on its back stroke, as usual, and it is fired by an electrical igniter. As the piston travels back, compressing the charge in the cylinder, it draws a fresh charge into the front end, 3, of the cylinder and the cross-head plunger of the other half of the unit similarly draws in a charge of air. The delivery of air from the chamber and passages 7, 20, and 19 is controlled entirely by the valve 11, but the intake of air by the plunger 6 is controlled by a piston valve 63, Fig. 12–31, which takes air from the chamber 68, connecting with the atmosphere through the base of the engine, and delivers it to the chamber 69, which is connected to the cross-head cylinder of the other half of the engine. The piston valve 62 takes in a mixture of gas and air through the chamber 67, which is connected with the supply source, and delivers it to the fuel pump 3, Fig. 12–30, through a balanced throttle valve 60, Fig. 12–31. The fuel pump then forces the mixture back through the throttle valve 60 to the admission valve cage. The throttle valve thus regulates both the quantity of mixture drawn in by the pump and the quantity delivered by the pump to the combustion chamber. The mixture and air-intake valves are operated by connecting rods from a rock-shaft, as indicated in Fig. 12–31; the rod 38 actuates the corresponding valves for the other half of the engine. This rock-shaft is oscillated by an eccentric on the main shaft and an eccentric rod.

The governor is of the fly-ball spring-opposed type and serves merely to control the position of the balanced throttle valves in the fuel passages. Since the cylinder is filled with scavenging air every stroke, the variation of the amount of mixture admitted does not vary the compression pressure but merely varies the richness of the cylinder contents.

The engine is equipped with both make-and-break and jump-spark igniters, but the former are ordinarily used.

The Fairbanks Engine. — The Fairbanks improved horizontal enigne is of the four-cycle type and made to operate on gas or on gasoline, distillate or alcohol.

The general appearance of the engine is well shown in Fig.

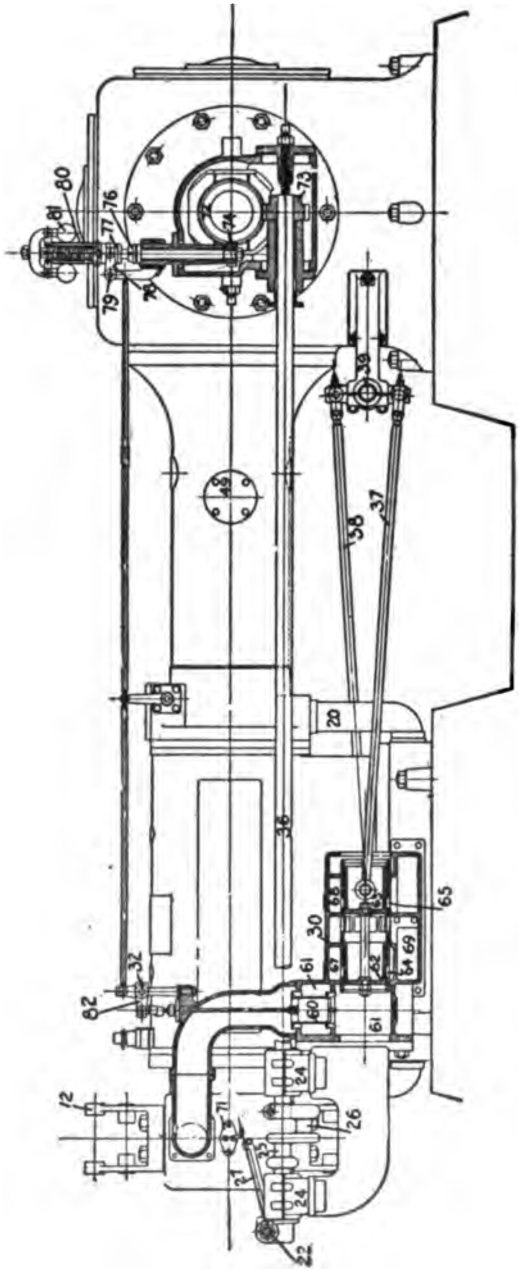


Fig. 12–31. — Governing Details of Buckeye Two-cycle Engine.

12–32 which shows the gas engine from the governor side, while Fig. 12–33 gives a front view of the same machine. The cylinder is partly supported by the frame, as shown. Cylinder jacket wall and cylinder head are cast in one piece, doing away with all joints. Both valves of the poppet type work upward and are mechanically operated. Fig. 12–34 shows the position of these valves at the head of the cylinder. Each valve is placed in a separate cage, and, when closed, the valve face is practically flush with the wall of the combustion chamber. This results in a combustion chamber of very simple form. The lay shaft at the side of the engine is driven by two-to-one gearing from the crank shaft. It carries, Fig. 12–32, first the bevel gear for operating the hit-and-miss fly-ball governor; second a clutch arrangement for making and breaking the connection between the inlet and igniter cam sleeve and the lay shaft; third, the exhaust valve cam, and lastly the sleeve carrying the inlet valve and igniter cams. The governor acts by interposing a pick blade, prevents the ex-

Fig. 12-32. — Fairbanks Engine.

Fig. 12-33. — Fairbanks Engine.

haust valve lever from returning, and thus blocks the exhaust valve open. In this position a projection on the exhaust valve lever unlocks the clutch above mentioned, breaks the connection between lay shaft and inlet cam sleeve, causing the latter to remain stationary. The inlet valve then fails to open as long as the governor does not withdraw the blade. It is possible to change the speed of the engine through a certain range by adjusting the governor during operation.

The ignition system is of the make-and-break type, as is clearly indicated in Fig. 12–33. The electrodes are contained in one block, which is easily removable for inspection.

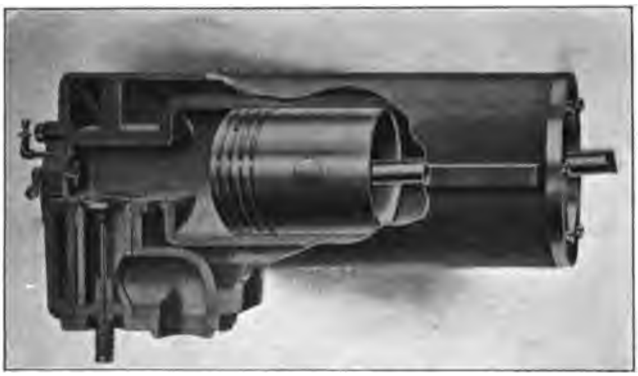


FIG. 12–34. — Cylinder Construction, Fairbanks Engine.

The method of operating the gas valve by means of the main inlet valve lever is also shown in Fig. 12–33.

The Fairbanks gasoline engine is in all respects similar to the gas engine above described, except that the mixing valve shown at the left of Fig. 12–33 is replaced by a simple type of overflow carbureter which is supplied by a gasoline pump operated by a cam on the lay shaft.

The Philadelphia Otto Engine. — The general features of the design of this engine are shown in Figs. 12–35 and 12–36. These particular drawings refer to a 30 horse-power illuminating gas engine, but the same design is carried out in all horizontal engines from 5 up to and including 40 horse-power. Excellent features of the construction are the separate cylinder liner and the remov-

able valve cages for gas, inlet, and exhaust valves. Make-and-break electric ignition operated by a crank from the end of the lay shaft is used.

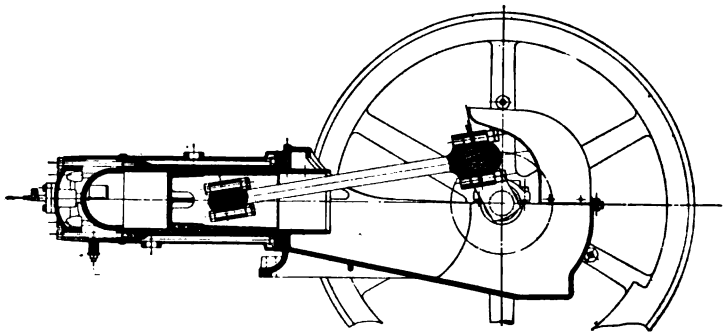


FIG. 12-35. — Philadelphia Otto Engine.

The producer gas engines made by this firm, the Otto Gas Engine Works of Philadelphia, show the same general make-up. Fig. 12-37 gives a general view of a type made in 60, 75, 95, and 120 horse-power sizes. This design differs from that of Fig. 12-35 mainly in that the cylinder is supported also near the end by an extension of the frame.

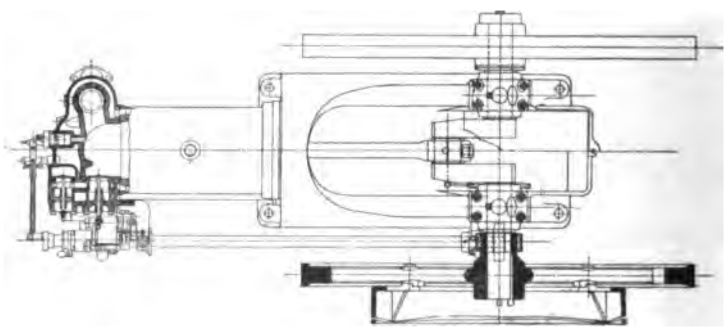


FIG. 12-36. — Philadelphia Otto Engine.

The suction gas engines are governed by controlling the fuel valve, regulation thus being effected by changing the quality of the mixture.

The Olds Gas Engine. — The Olds Gas Power Company of Lansing, Mich., build two types of gas engines: Type G from

8 to 100 horse-power and Type K from 25 to 300 horse-power, both being horizontal single-acting four-cycle engines.

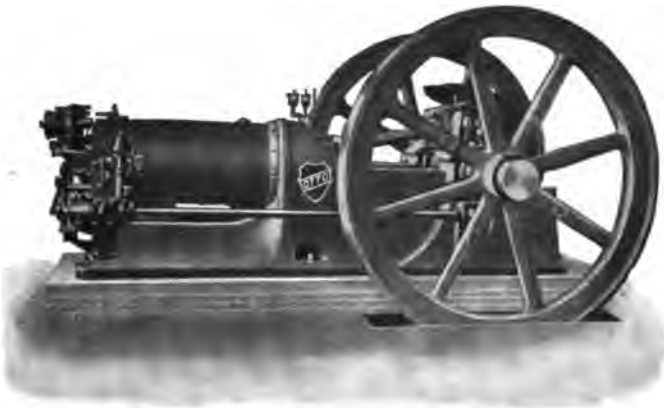


FIG. 12-37. — Philadelphia Otto Producer-gas Engine.

Type G is illustrated in Fig. 12-38. This engine may be used either for gas or gasoline. There appear to be no very unusual features in its general design. Both inlet and outlet valves are

FIG. 12-38. — Olds Type G Engine.

of the poppet type, but an auxiliary exhaust opening is used to relieve the main exhaust valve. The exhaust valve is mechani-

cally operated by means of a straight push rod and cam. Ignition is by make and break. The governor is of the hit-and-miss type and operates to hold the exhaust valve open. The mixing arrangements are not specially described in the available information on this engine.

Type K engines are of somewhat different design and embody in their make-up the best and most advanced ideas. A general view of the machine is given in Fig. 12–39. Among the excellent features of this design are the following: The jacket wall is integral with the frame. The cylinder liner is made of a grade of

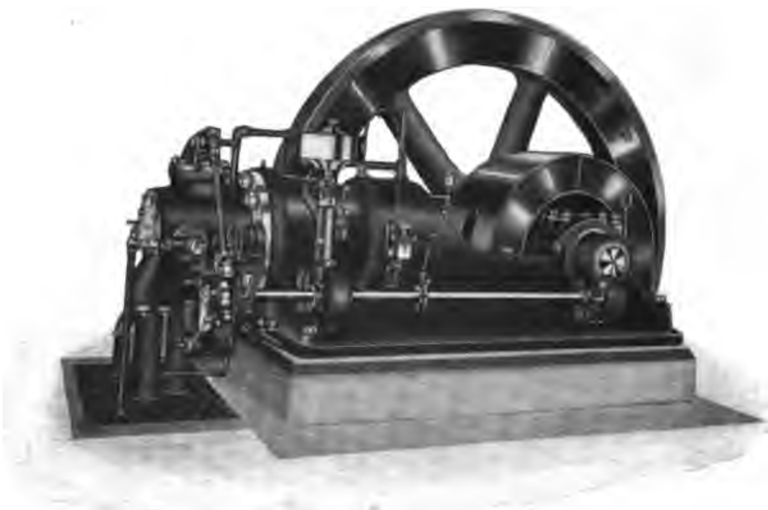


FIG. 12–39. — Olds Type K Engine.

metal especially adapted to the service and consists of a straight cylinder with a flange at the outer end. This flange is received into the frame and is held in place by the cylinder head. This construction allows of even and unrestricted expansion. The cylinder head contains the openings for the inlet and outlet valve cages and is designed with the greatest possible regard to expansion and cooling stresses. The following description of the valve mechanism is taken from the catalogue published by the company.

The inlet and exhaust valves are of the vertical poppet type,

mechanically operated and working in long guides in the same vertical axis with the inlet valve at the top and the exhaust valve at the bottom. The inlet valve and gas valve have a common stem and the cage is so arranged that a thorough mixing occurs just at the entrance to the cylinder. The gas valve opens slightly later than the air valve and by special construction used a perfect seating of both valves is assured at all times. On the smaller sizes, by removing the inlet valve cage the exhaust valve is perfectly accessible, while on the larger sizes the exhaust valve may be removed together with its hollow water-cooled seat without disturbing the inlet valve. This is easily done as its weight is counterbalanced. Both valves are operated by a single cam which is designed so as to have a quick, full valve opening, without noise or clatter. Valves and valve ports are of liberal sizes so undue throttling is avoided. The valve springs are of the highest grade spring steel, and rest on plates which, being supported on ball and socket joints, do away with side thrust. All parts and particularly all bearings are made of ample dimensions with provision for oiling every wearing surface. The lay shaft which is driven from the crank shaft through spiral gears operates the valve mechanism, governor and ignition mechanism.

The speed is controlled by a governor, apparently of the Hartung type, which is driven from the lay shaft and serves to throttle the mixture admitted to the cylinder.

Figure 12–40 gives a good view of the ignition arrangements. The current is supplied by a low-tension make-and-break Bosch magneto, the operation of which is explained in Chapter XIII. The point of ignition can be easily changed during operation by adjusting the lever along the row of holes *A–R*.

The Warren Engines. — The Struthers-Wells Company of Warren, Pa., manufacture various types of engines, both hit-and-miss and throttling, all operating on the four-cycle principle.

For ordinary power purposes the firm builds the single-cylinder hit-and-miss engine illustrated in Fig. 12–41. This engine is made in sizes from 10 to 90 horse-power. Above 30 H.P. the installations are equipped with special starting devices.

All of the other types are apparently governed by throttling. The next higher range of power, 35 to 200 horse-power, is covered

by the two-cylinder throttling engine shown in Fig. 12–42. The manner of governing is clearly indicated.

A special type of the two-cylinder engine, for which maximum economy and closest regulation is claimed, is shown in elevation in Fig. 12–43. This engine is made in two sizes only, 110 and 125 horse-power. From the vertical section, Fig. 12–44, showing the cylinder construction, it is seen that the cylinder barrel and jacket are cast in one piece. The valves are of the vertical poppet

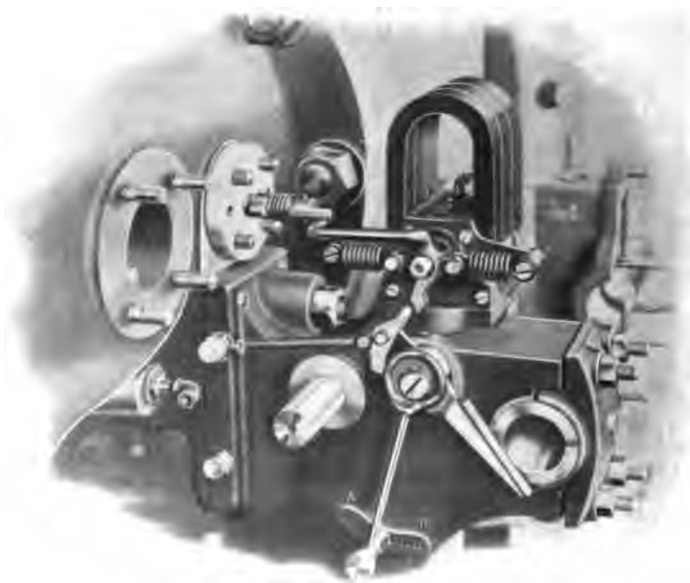


FIG. 12–40. — Igniter Details, Olds Type K Engine.

type, opening upward, and are placed in a valve cage integral with the cylinder head. The manner of operating these valves by cams and levers from a valve shaft running across the engine under the cylinders is shown in both Figs. 12–43 and 12–44. Governing is effected by throttling the mixture in the supply pipe just before it divides. The centrifugal governor and its linkage are indicated in Fig. 12–45. In some cases the governor is located between the cylinders as shown in Fig. 12–43. A cross-section through the valve chest, Fig. 12–46, shows the method of operat-

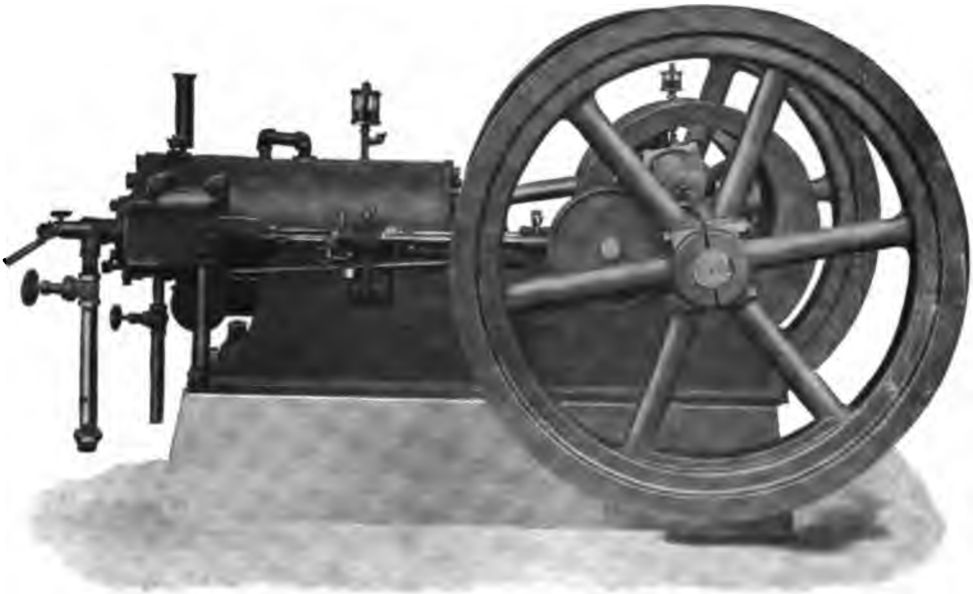


Fig. 12-41.

Fig. 12-42. — Warren Throttling Engine.

FIG. 12-43. — Warren Throttling Engine.

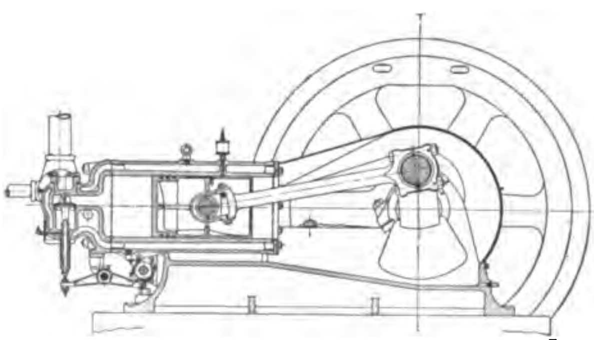


FIG. 12-44. — Vertical Section Warren Throttling Engine.

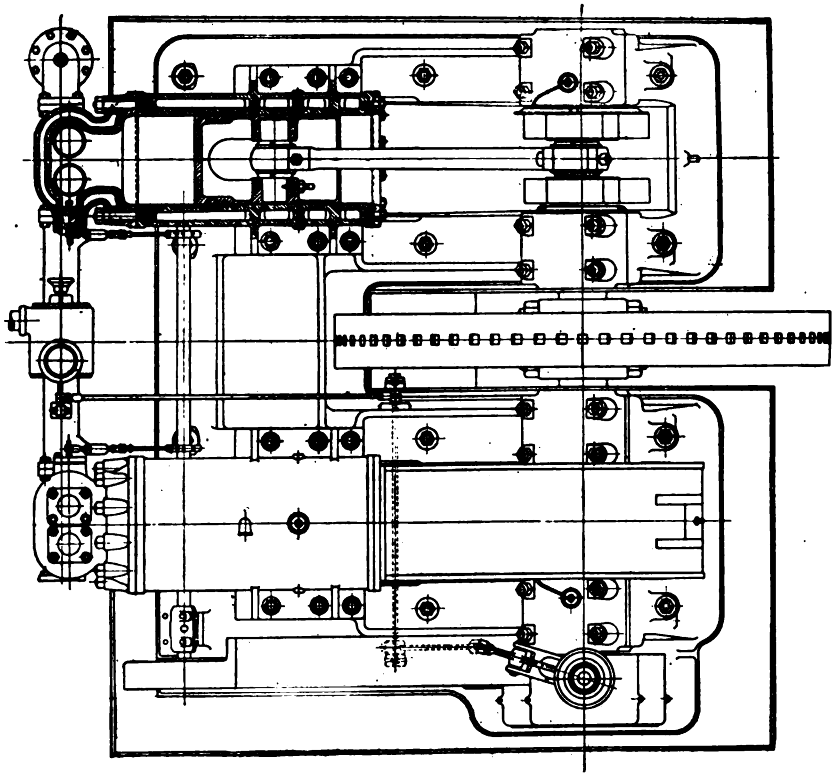


FIG. 12-45. — Plan of Warren Two-cylinder Engine.

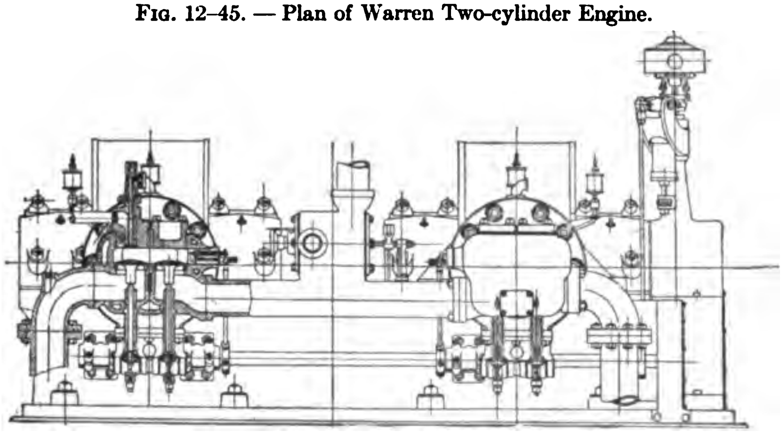


FIG. 12-46. — Cross-section through Valve Chest, Warren Two-cylinder Engine.

ing the make-and-break igniter, and the location of the starting valve in the valve chest cover just above the exhaust valve.

Warren engines, covering the range from 200 to 325 horse-power, are of the tandem single-acting type, a general view of which is shown in Fig. 12–47. The cross-section, Fig. 12–48, shows the cylinder construction, which is unique in some of its features. The back part of the main frame forms the jacket wall for the front cylinder. The cylinder barrel itself is cast in one piece with the cylinder head. The construction of the back cylinder is similar. The distance piece between the two cylinders rests on a separate base, and forms, for a portion of its length, the jacket wall for the rear cylinder. The front piston is of the ordinary trunk type. The back piston is longer than is usual in pistons not subject to side thrust. The method of water-cooling piston and rod is clearly shown.

The valves are of the vertical poppet type held in separate cages which are easily removable. The exhaust valves, at the bottom of the cylinder, are water-cooled. The valve gear is shown in detail in Fig. 12–49. Both valves are operated from the same cam on the lay shaft. The operation of the exhaust valve is clear. The motion of the inlet valve varies, depending upon the load. The valve opens and closes always at the same time, because the lift of the actuating cam is not changed throughout the entire range of load. The governor, however, through the linkage shown, controls the position of the sliding block above the valve lever, moving it in or out, depending upon whether the load falls or rises. This block acts as the fulcrum about which the valve lever turns, and hence the lift of the valve is made proportional to the load. The inlet valve stem carries the gas valve. The latter opens somewhat later than the main inlet valve. The manner of mixing gas and air is clearly indicated in the figure. Butterfly valves in the air and gas passages serve to help control the proportions of the mixture. The ignition system, operated from the lay shaft as shown in Fig. 12–49, is of the make-and-break type.

The Struthers-Wells Company also builds vertical engines up to 600 horse-power. The older multi-cylinder types of these machines are practically nothing but two or three separate engines direct connected. Thus a 450 horse-power unit now in operation consists of three engines with two fly-wheels between the cylinders.

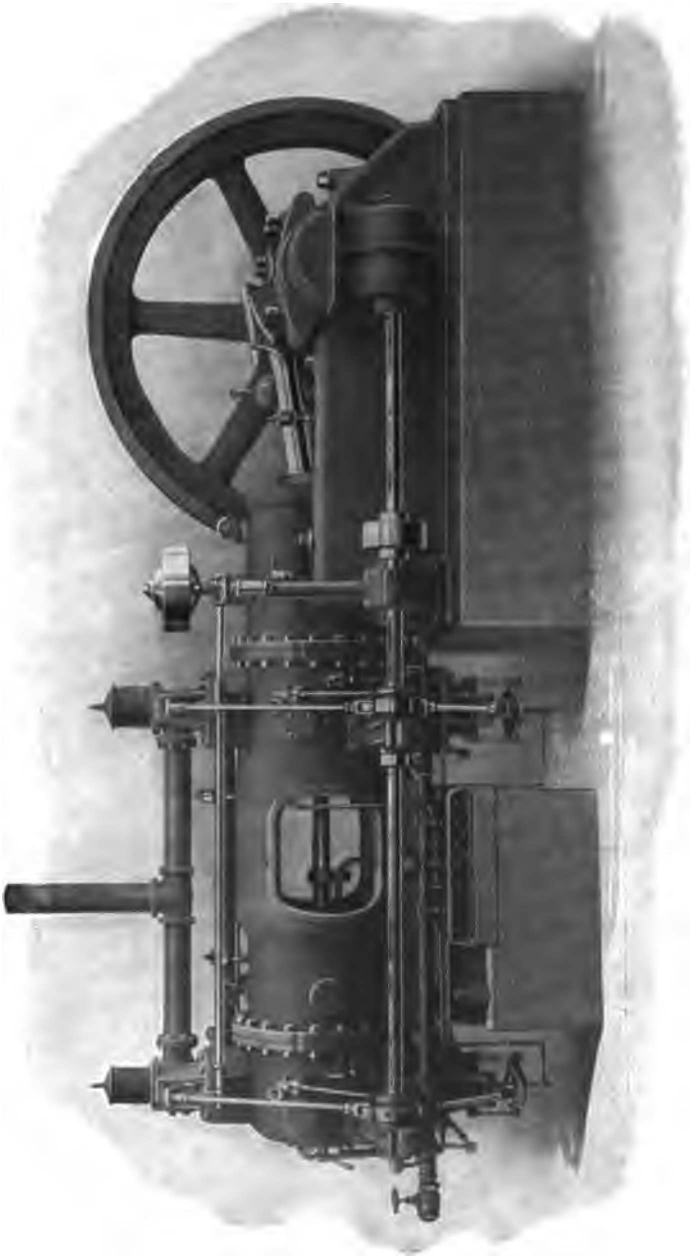


Fig. 12–47.

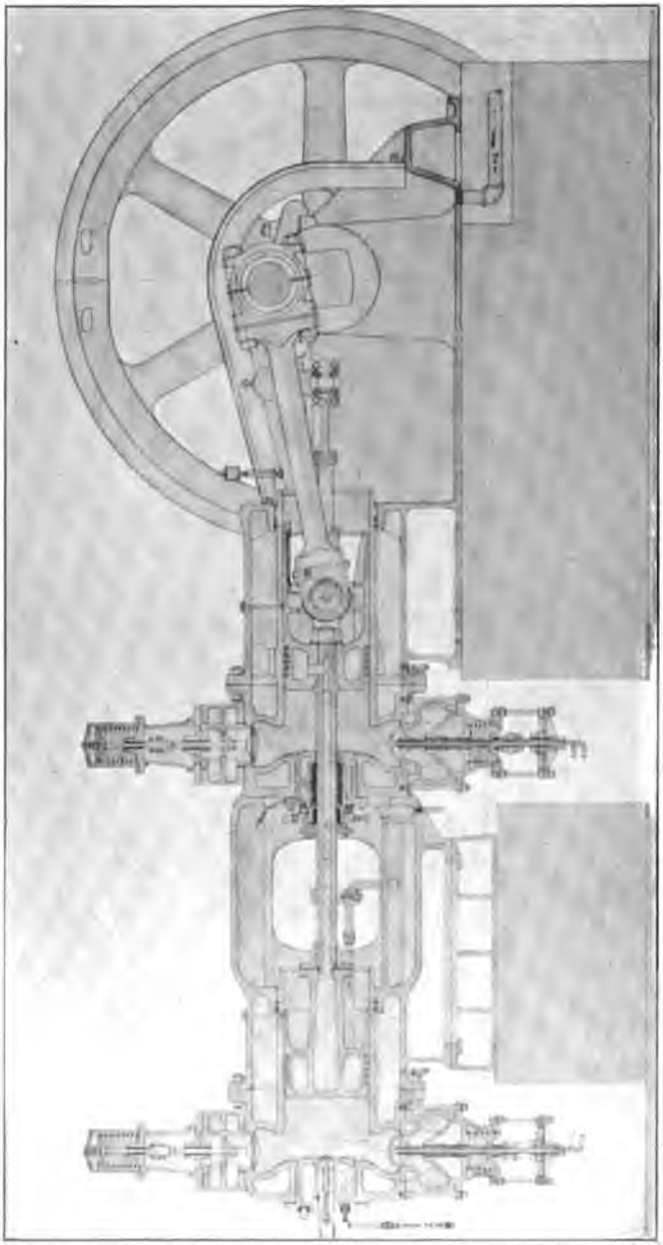


FIG. 12–48. — Cross-section of Warren Single-acting Tandem Engine.

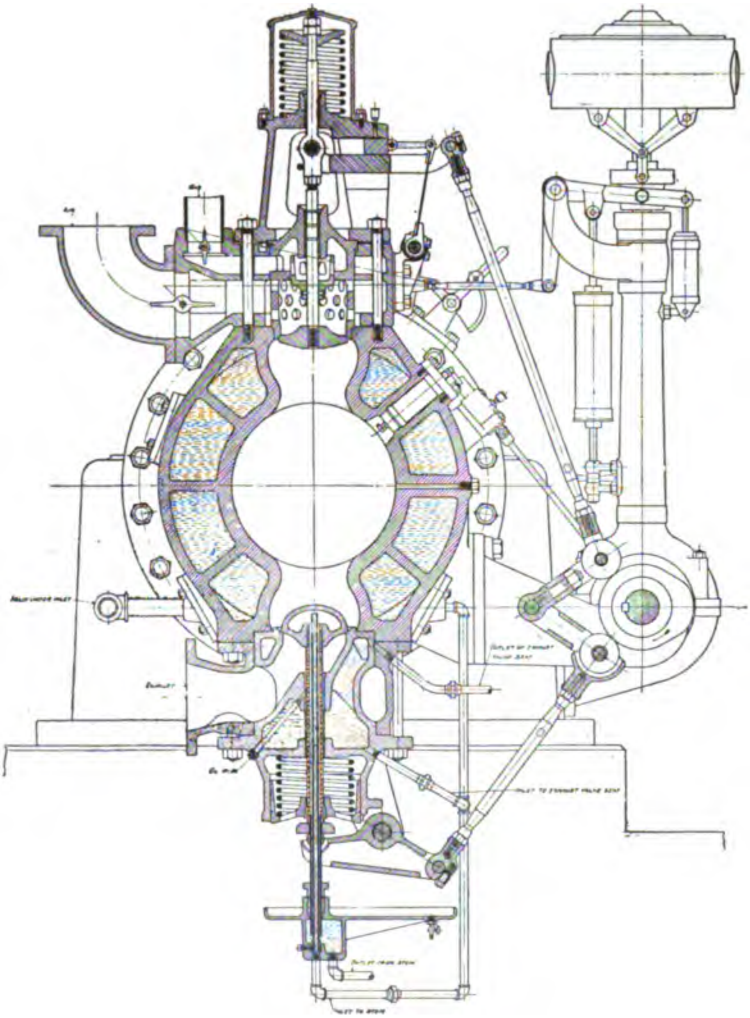


Fig. 12–49. — Valve Gear of Warren Single-acting Tandem Engine.

In the later designs this has been modified. A 600 horse-power vertical engine lately completed is of the four-cylinder A-frame type mounted on a solid bedplate with the fly-wheels at the ends.

2. LARGE GAS ENGINES. — Large gas engines of American design are manufactured by the Westinghouse Machine Company, by the William Tod Company, Youngstown, Ohio, by the Snow Steam Pump Company, Buffalo, by the Riverside Engine Company, Oil City, Pa., and by the Wisconsin Engine Co., of Corliss, Wis., makers of the Sargent engine. There are, however, a few other firms making large engines of foreign design. Thus the De La Vergne Machine Company of New York make the Koerting two-cycle engine, and the Power and Mining Machinery Company the Crossley engine. The Allis-Chalmers Company, who built the Nürnberg engine, have apparently an engine on the market that is not strictly of Nürnberg design. Besides these well-known machines made in this country, the Premier engine made in England, the Cockerill engine made in Belgium, and the German Oechelhäuser and Deutz engines, should be mentioned. The Oechelhäuser and Koerting engines are made under license by a number of firms in Germany, and in the case of the Koerting engine, also abroad. While it has in general been easy to get sufficient descriptive material on the above-mentioned engines, this does not apply to certain machines of American design, and the information given is hence somewhat meager.

The Westinghouse Horizontal Engine. — There seems to be available practically no definite information on the constructive details of the Westinghouse horizontal double-acting tandem engine.* The older type apparently used the center crank and the combustion, or at least the valve, chambers were placed at the sides of the cylinders. The later types of horizontal engines built by this company approach much nearer to established European practice. Thus the engines installed in the power plant of the Warren and Jamestown railway system have the valves in the central lines of the cylinders, as shown in Fig. 12–50. The latest design is shown in Fig. 12–51, which represents a 3000 horse-power unit for the power plant of the Carnegie Steel Company at Bessemer, Pa. In this type the side crank is used, but outside of this

* A full description of the Westinghouse Engine has just appeared in Power, April, 1908.

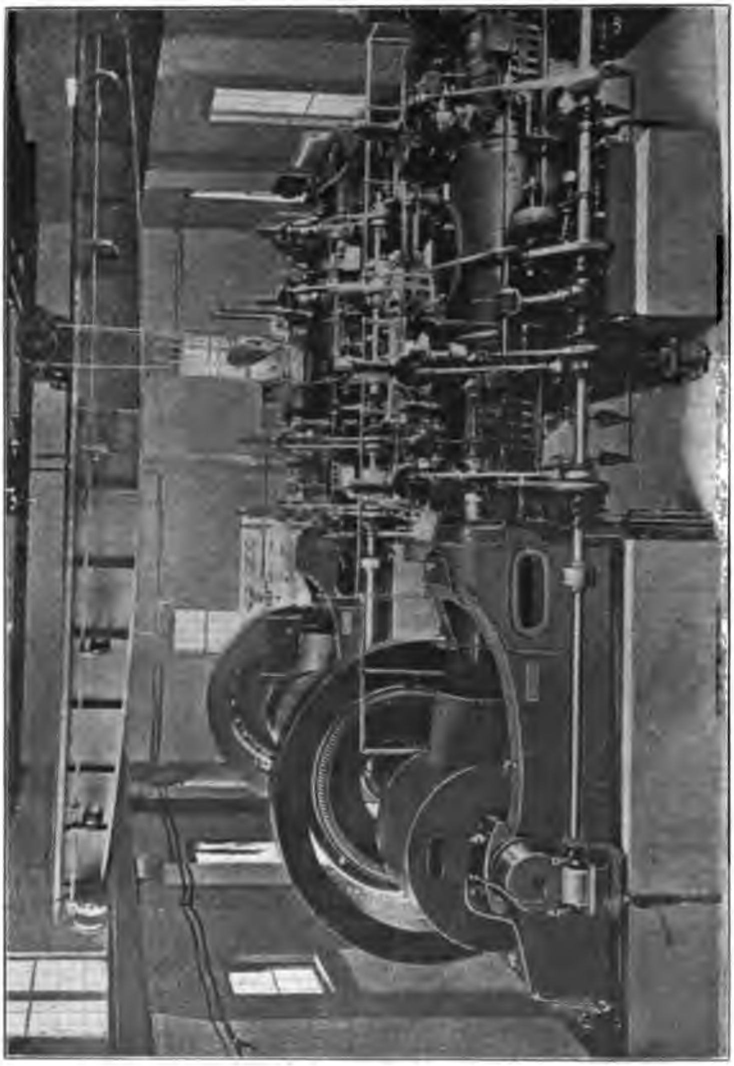


Fig. 12-50. — Westinghouse Double-acting Tandem Gas Engines.

nothing is definitely known to the writer regarding the mechanical details. An interesting development of very recent date is the

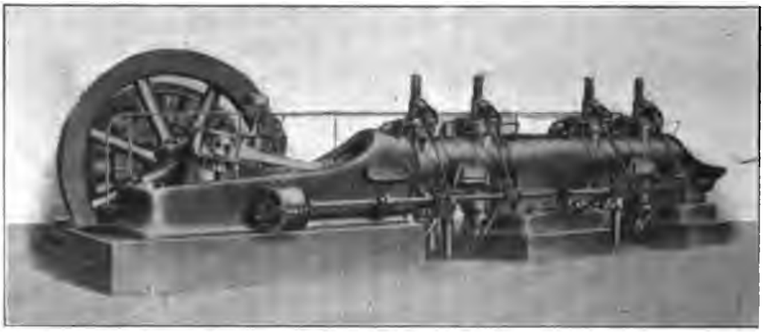


FIG. 12-51. — Westinghouse Double-acting Tandem Engine.

Westinghouse vertical single-acting tandem engine shown in Fig. 12–52. This engine was built by the British Westinghouse Electric and Manufacturing Company.*

The Tod Engine. — The following description of this engine is taken from the *Iron Age* of July 18, 1907:

FIG. 12-52. — Westinghouse Vertical Single-acting Tandem Gas Engine.

"The general appearance of the engine is shown in Figs. 12–53 and 12–54. The cylinders are arranged in pairs, two cylinders being connected in tandem to each crank pin. The valve gear, igniter, switchboard and operator's hand wheels are all situated between the cylinders and are easily accessible from an operating platform placed just below the level of the center lines of the cyl-

* W. H. Booth, Cassier's Magazine, November, 1907.

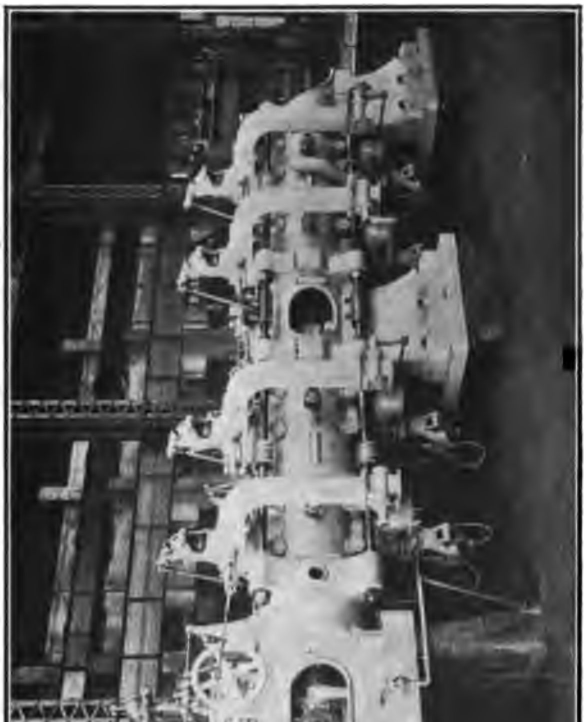


FIG. 12-53. — Tod Double-acting Twin Tandem Engine.

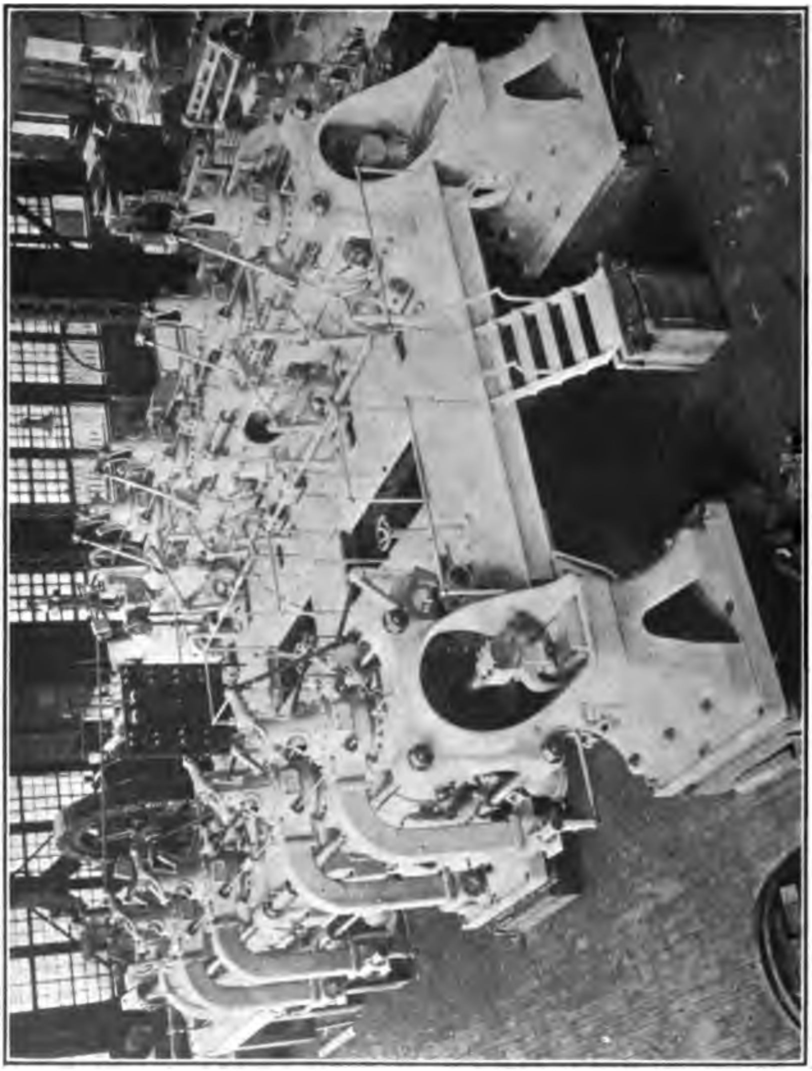


Fig. 12-54. — Tod Double-acting Twin Tandem Engine.

inders and shaft. The foundation level under the cylinders being several feet lower than under the frames, leaves ample room below the cylinders to make the exhaust valves, etc., easily accessible, a very desirable but somewhat unusual feature.

"The cylinders and water jackets are integral and cast in halves, secured together with flanged joints at the center. The cylinders are not attached directly to the main bedplates nor to each other, but are supported by the tie pieces in such a way that the barrels themselves are entirely free. All strains are transmitted through four heavy forged steel tie bolts extending the entire length of the cylinders, and attaching directly to heavy lugs on the bedplate. This obviates the transmission of the strains through the cylinder walls, and contributes to accessibility and the easy removing of parts. The pistons are of steel, with cast-iron junk rings, and the pistons and rods are water-jacketed in the usual manner. Adjustable tail-rod supports take the weight of the pistons and rods from the cylinders.

"The valve gear is driven by eccentrics, eliminating entirely the cam drive, which has been an objectionable feature of many of the gas engines heretofore built. There is one eccentric for each end of each cylinder, which drives both the inlet and exhaust valves — an arrangement that reduces the number of parts. The eccentrics are mounted on two lay shafts running parallel to the axes of the cylinders. The latter are driven by a cross shaft, which, in turn, derives its motion from two eccentrics mounted on the main shaft. The inlet valves are on top of the cylinders, and the exhaust valves on the bottom. The inlet valve proper is of the mushroom type, sealing the ports from the pressure in the cylinders. The main valve is operated by a rolling lever and returned to its seat by a spring. The mixing valves and governor valves are of radial gridiron type, and are located in the upper section of the valve bonnet. The mixing valves may be operated individually or collectively by a suitable hand mechanism. The governor valve which controls the admission of gas has constant travel, the time of opening being controlled by the governor by either increasing or decreasing the angle of advance of the crank which operates them, since the engine is of the constant-compression type.

"The governor is of the fly-ball pattern and is in duplicate,

one controlling the operating valves on each side of the engine. The two are driven by Morse chains through a flexible coupling, and are connected by a cross rod which may be removed in case it is desired to operate either side of the engine alone.

"Each end of each cylinder is equipped with two igniters, one operated mechanically and the other by a solenoid. Either may be used independently or the two together. The igniters are under control of the governor, and when the engine is at rest are automatically thrown back to the dead center. The ignition is on the make-and-break system, using direct current at 90 volts, supplied by a motor generator set, which is so designed that either end may be used as a motor or a generator, or both may be used as generators by driving directly from the engine shaft. Connected in series with each igniter is a tell-tale lamp on the switchboard, giving a positive indication as to whether or not the igniters are sparking, short-circuited or burnt out."

The Sargent Engine. — The Sargent complete expansion engine, made by the Wisconsin Engine Company of Corliss, Wis., is shown in general elevation in Fig. 12–55 and in transverse section in Fig. 12–56. As far as the constructive details of this engine are concerned, it has the merit of great simplicity. The engine is built as a double-acting tandem. There is but one valve to control admission and exhaust for each end of each cylinder, and but a single cam to perform these various offices, while a second cam operates the igniter. The lay shaft is driven from the main shaft by a pair of worm gears. The governor is of the inertia type, the Rites, and operates to advance or retard the lay-shaft, thus controlling the time of cutting off the admission of the incoming charge to the cylinder. This construction is decidedly different from that used by other designers.

The combination valve is shown in cross-section in Fig. 12–56. Its operation is described as follows:

"Gas is piped to the chamber *A* in the sub-base and air to the chamber *B*, which pass through the cylinder supports to the chambers *A′* and *B′*, ready to pass into the mixing chamber when the cam depression *M N* passes the roller and the ports *F* in the piston valve register with the ports *E* and *D* in the bushing. When the piston valve goes down to this position, the confined air in the piston valve dash-pot forces open the poppet valve,

FIG. 12-55. — Sargent Complete Expansion Engine.

thus giving free admission to the charge. When the point N of the cam reaches the roller, it is forced down, while the other end of the lever goes up, carrying the piston valve which cuts off the admission. The poppet valve seats and both valves remain

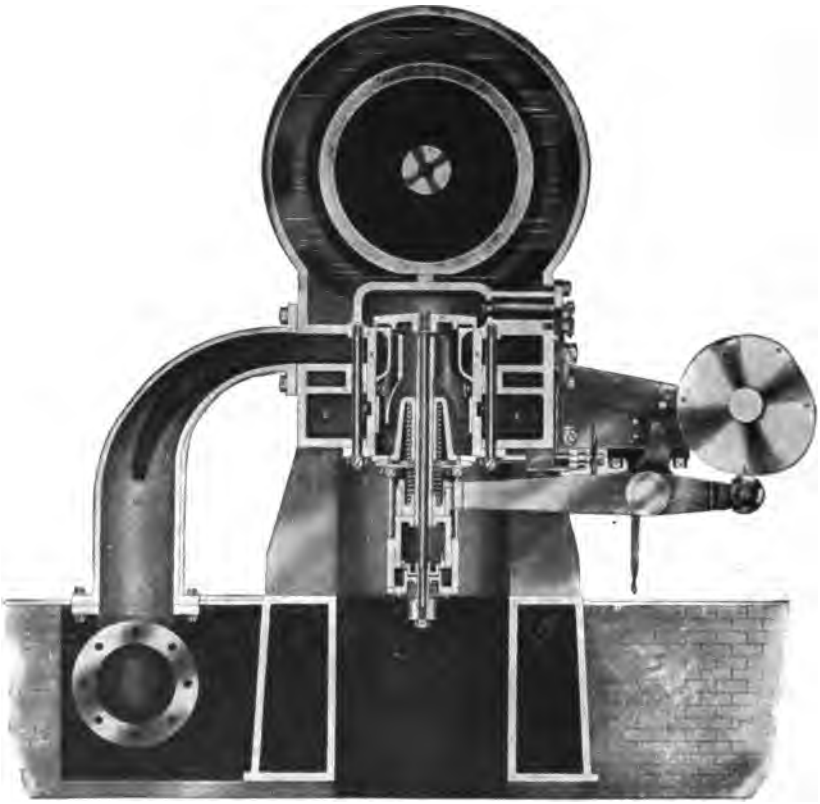


FIG. 12-56. — Transverse Cross-section Sargent Engine.

in normal position during compression, ignition, and expansion, or until the point L on the cam pushes the roller down and the piston up, which opens the poppet and the exhaust gases pass out through the ports K and the elbow W to the exhaust pipe under the floor.

"The poppet valve seals the opening in the combustion chamber during the compression and inflammation, and the piston valve,

holding against no pressure, works loosely in its bushing, cutting off the admission and guiding the exhaust.

"As the poppet valve controls both the inlet and outlet gases, both the valve and seat keep cool and need to be ground not over once or twice a year. The mechanism is simple and, as the roller is always bearing on the cam, the valve motion is practically noiseless."

By revolving the piston valve by the index wheel, the blind port *S* varies the mixture to suit the gas whether it has 100 or 1000 B. T. U. per cubic foot.

The fundamental idea of the Sargent engine is to get complete expansion of the charge in the cylinder. In the ordinary engine, which at full load draws in the charge to the full stroke of the piston, or nearly so, the terminal pressure is anywhere from 25 to 50 pounds, and the final temperature over 1000 degrees Fahrenheit. Sargent claims that his engine, cutting off the stroke at full load at three-quarters of the suction stroke, will show a terminal pressure slightly above atmosphere and a temperature of about 400 degrees Fahrenheit. The system of speed regulation by cutting off at various points along the suction stroke is one used by several makers, but in nearly all cases the engines cut off at nearly full stroke for full load, that is, at full load the ratio of compression is nearly equal to the ratio of expansion. In the Sargent engine the ratio of expansion always exceeds the ratio of compression, and a lowering of the terminal pressure and temperature is of course the result. There is no doubt that this lowering of the temperature has a great deal to do with any success that the use of a combination valve such as described above may have. In the Sargent engine the rated power is developed when the governor cuts the admission off at three-quarter stroke; the rest of the stroke may be considered potential over-capacity. In spite of the fact that ordinary experience shows a gas engine to be most efficient when it is developing its maximum, not the rated, load as long as normal speed is maintained, the claim is made for this engine that its best efficiency is obtained when cutting off at three-quarter stroke.

The Koerting Two-Cycle Engine. — The Koerting engine is of German design, made in this country by the De La Vergne Machine Company of New York. This is one of the two successful large

two-cycle gas engines in Europe, the other being the Oechelhäuser, not made in America. Koerting two-cycle engines are now made by several German firms, whose constructions differ somewhat among themselves and from the parent design, mainly in frame and governor details. The latter seem to have an important bearing upon the amount of work done by the pumps. The type made by the De La Vergne Machine Company is best illustrated from the catalogue of the firm. Fig. 12–57 shows a cross-sectional elevation and a plan of this machine, used in this case as a blowing engine. It is seen that the engine has but a single cylinder which is double-acting. There are thus exactly the same number of power impulses per turn as in a single-cylinder steam engine. The piston is very long, about seven-eighths of the stroke, and is of course water-cooled. This makes it necessary to support its weight at the cross-head and at a tail-bearing to save both cylinder and stuffing-boxes. At the side of the power cylinder there are a gas and an air pump which furnish gas and air by separate passages to the mixture inlet valves at the top of the cylinder. The manner of driving these pumps from a side crank and of operating the piston valves, which control them, by means of rocker arms driven by an eccentric from the main shaft, is clearly shown in Fig. 12–58. Fig. 12–59 illustrates the valve shaft side of a partially constructed machine and shows the manner of operating the inlet valves and igniter gear. The exhaust gases are taken care of by a ring of ports in the middle of the cylinder. These ports are uncovered by the piston alternately on each side (see Fig. 12–57).

The operation of the engine is as follows:

It will be seen from Fig. 12–58 that the pump crank is in the neighborhood of 100 degrees ahead of the main crank, *i.e.*, when the latter is at either dead center the pumps have completed about one-half their stroke. From the beginning of its discharge stroke the air pump B, Fig. 12–60, has forced its charge of air into the passage leading to the main inlet valve and has forced some of this air also partly up to the gas passage, which is the inner concentric passage surrounding the inlet valve stem. In the meantime the piston of the gas pump A, the motion of which has been exactly the same as that of the air pump piston, has been allowed to shove back part of its charge into the gas suction pipe.

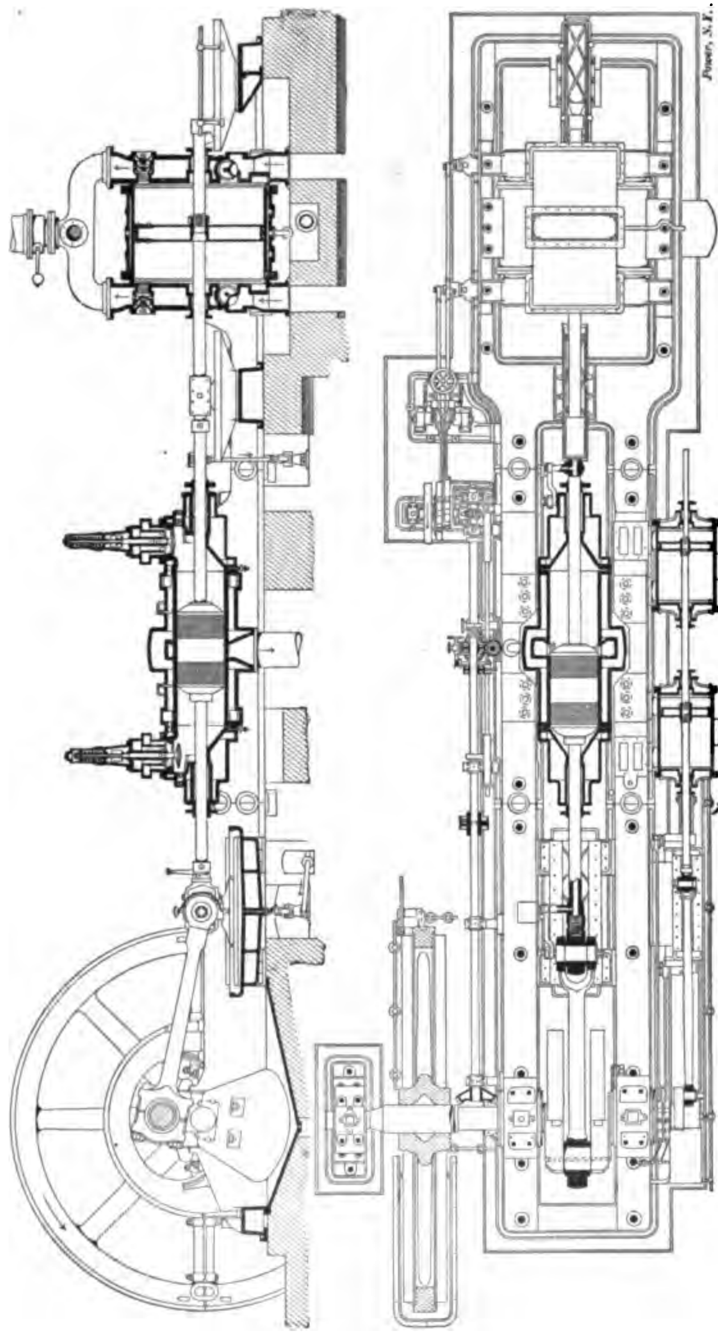


FIG. 12–57. — Koerting Two-cycle Engine, for Blowing Service.

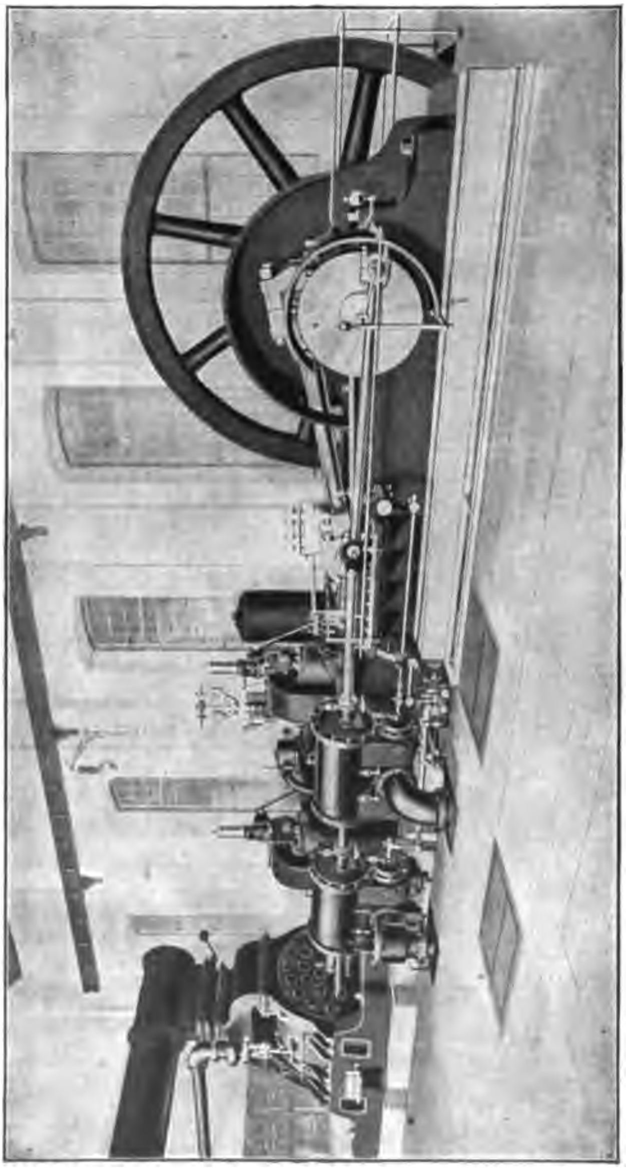


Fig. 12-58. — Pump Side of Koerting Two-cycle Engine.

At the moment, therefore, when the exhaust gases have nearly completely escaped from the main cylinder through the ring of ports, and the main inlet valve is opened, there is at first a rush of air from both the air and gas passages. This serves to drive the remainder of the exhaust gases out of the cylinder. A moment later the gas pump starts to deliver gas and the mixture enters the cylinder. By the time the power piston has covered the exhaust ports on its return stroke, the air and gas pistons

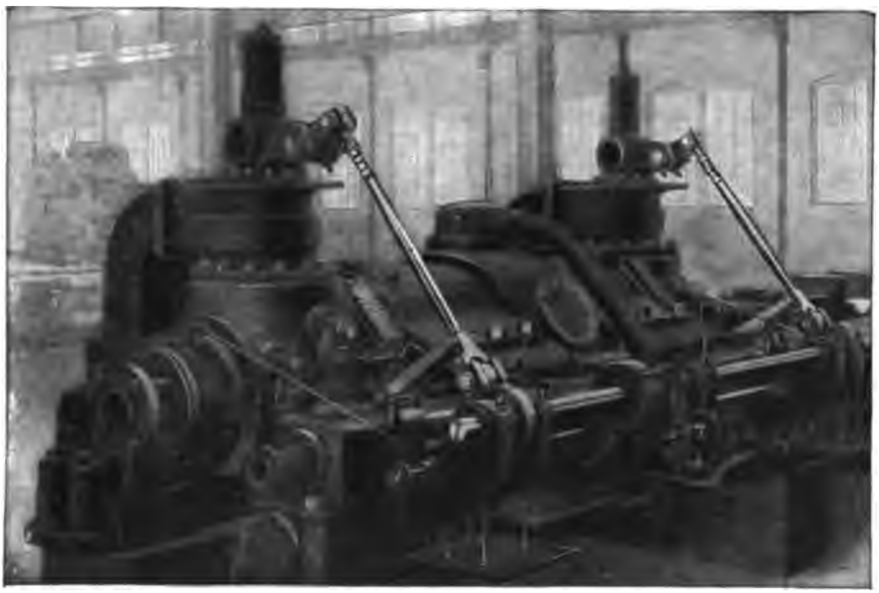


FIG. 12–59. — Valve Gear of Koerting Two-cycle Engine.

have reached the end of their stroke, the inlet valve closes and the mixture is compressed in the power cylinder. Ignition is by electric spark.

The speed of Koerting two-cycle engines is controlled by proportioning the amount of gas to the load. In the American type this is accomplished by putting butterfly throttle valves *f–f*, Fig. 12–60, in the discharge passages of the gas pump. The position of these valves is controlled by the governor, which, as some of the other illustrations show, is of the centrifugal type. This method of pump control is open to the objection that if the

gas used is at all dirty, the frictional resistances soon become so great as to render the governor inoperative. Some of the improvements of the later Koerting engines have been devoted to this very point. A modified design of gas pump is that of Klein Bros., Dahlbruch, Fig. 12-61.* In this construction the piston forces the charge back into the suction space through the ports shown at the middle of the cylinder, for one-half the stroke. The amount of gas delivered to the discharge passage during the last

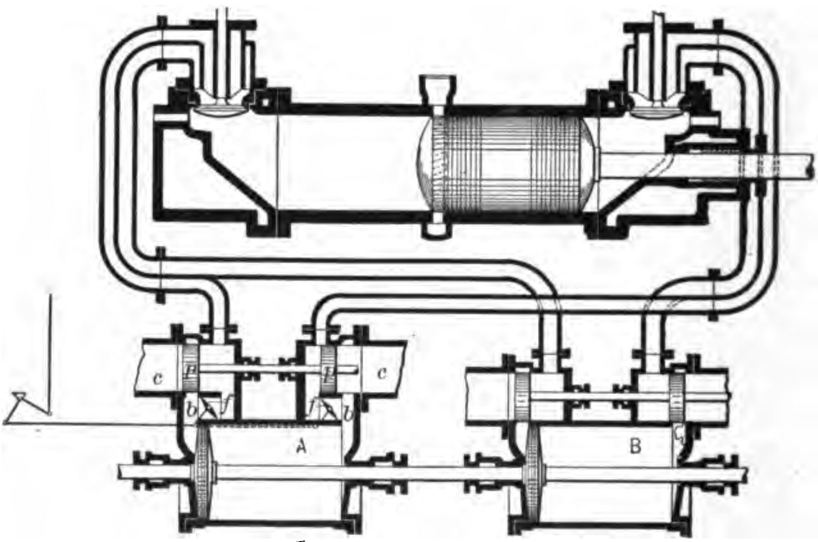


FIG. 12-60.

half depends upon the position of the small overflow piston valves near the bottom. This is controlled by the governor through the linkage shown. Another point that has given some trouble in the first designs of Koerting engines is the fact that the time available for the opening and closing of the inlet valve was but a very small part of the time of one revolution. The inertia actions were very severe and hence the engine speed was somewhat restricted. This difficulty has been overcome in an elegant way in the design of the Siegener Maschinensban *A-G* as shown in Fig. 12-62. Here the lay shaft moves only at one-half the speed of

* Hoffman, Zeitschrift d. V. d. I., September 15, 1906.

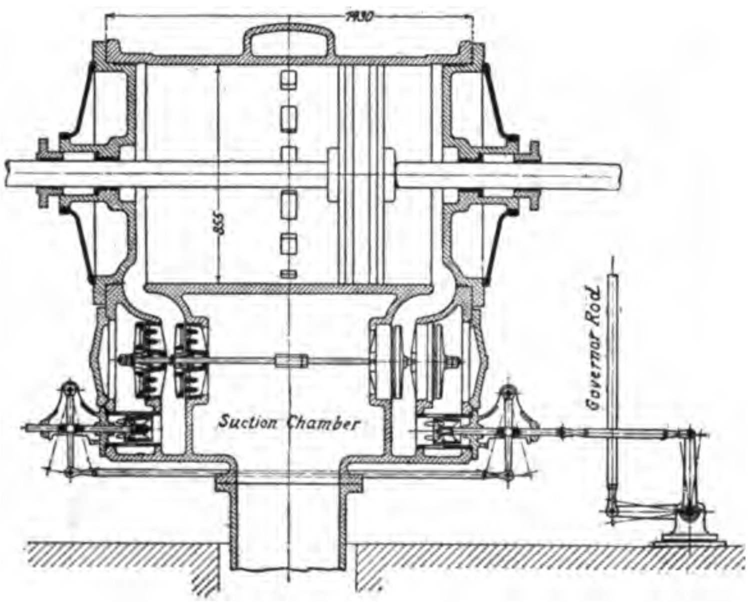


FIG. 12-61. — Gas Pump, Koerting Two-cycle Engine, Klein Bros.

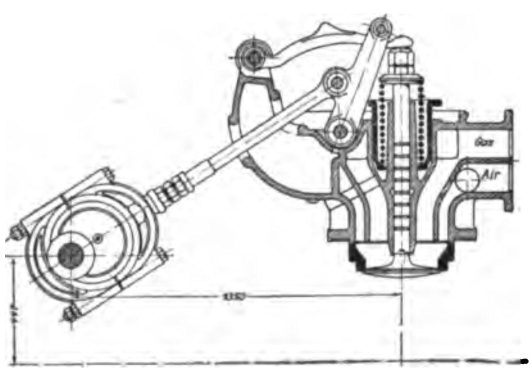


FIG. 12-62. — Modification of Inlet Valve Gear, Koerting Two-cycle Engine.

the main shaft, the same as in four-cycle engines, but the valve is opened twice for every turn of the shaft.

Reinhardt states that the Koerting engine is well adapted to blowing service because it starts easily under load and is certain in its operation through a wide range of speeds.

The Riverside Engine. — This engine is made by the Riverside Engine Company of Oil City, Pa., and embodies in its details several features decidedly different from those of other large engines. Fig. 12–63 shows the heavy duty double-acting tandem type and illustrates the peculiarity of the valve construction very clearly. Besides this type, this firm also builds single-acting,

FIG. 12–63. — Riverside Heavy Duty Double-acting Tandem Gas Engine.

single-cylinder and twin engines, and single-acting tandem engines. One of the single-acting, single-cylinder engines is illustrated in Fig. 12–64. The following is a description of the main features of the double-acting machine. It applies with little modification also to the single-acting type. Fig. 12–65 shows in cross-section the cylinder, valve and piston details of the double-acting engine and will serve as an aid in following the description of these parts.

The main frame or bed is of the heavy duty tangye rolling-mill type with bored guides and main bearings cast in a single piece, and is of great weight and extreme rigidity. The cylinder end of main frame is squared similar to cross-section of cylinder, and has machined holes for receiving the tie bars which attach the cylinder. The design of this frame, with its bored guideway, is such that a large portion of the metal is above the center line, making it exceptionally stiff.

The shaft is of side crank, built-up type and is machined all over.

The cylinders are made in two halves, with heads and valve chest cast integral without joints or packing, and are held rigidly to the main frame by four heavy steel tie bars which take all tension strains. The cylinders are mounted on a heavy cast-iron sole plate, having a machined top surface which keeps cylinders in alignment, permits perfect freedom for expansion and contraction, and by removing distance piece, cylinders can be slid endwise on sole plate, giving easy access to interior of cylinder, pistons

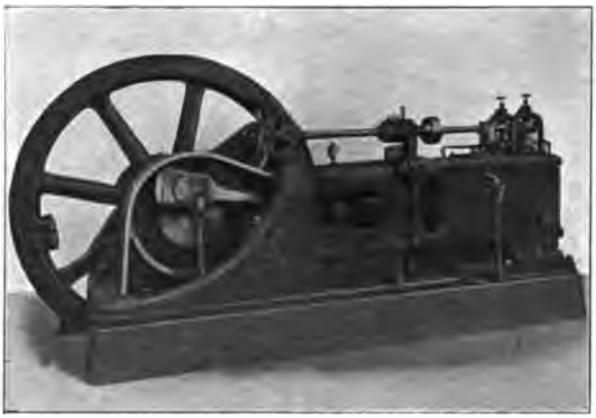


FIG. 12 64. — Riverside Single-acting Single Cylinder Gas Engine.

and piston rings. Rings can be cleaned or changed without disturbing piston or rod. The sole plate makes a drip pan under all the cylinders, keeping oil drip from foundation. The exhaust and inlet piping is attached to the sole plate, hence no piping except the water-jacket piping has to be disturbed when cylinders are moved. There is no overhead piping or wiring to interfere with traveling crane.

Both inlet and exhaust valves are of the semi-balanced water-cooled poppet type, operated in a vertical position. All water passing to the cylinder jacket passes through valves first, giving perfect and positive cooling without any attention whatever.

The balancing pistons run in renewable liners and are lubricated positively. These pistons form a large and perfect guide,

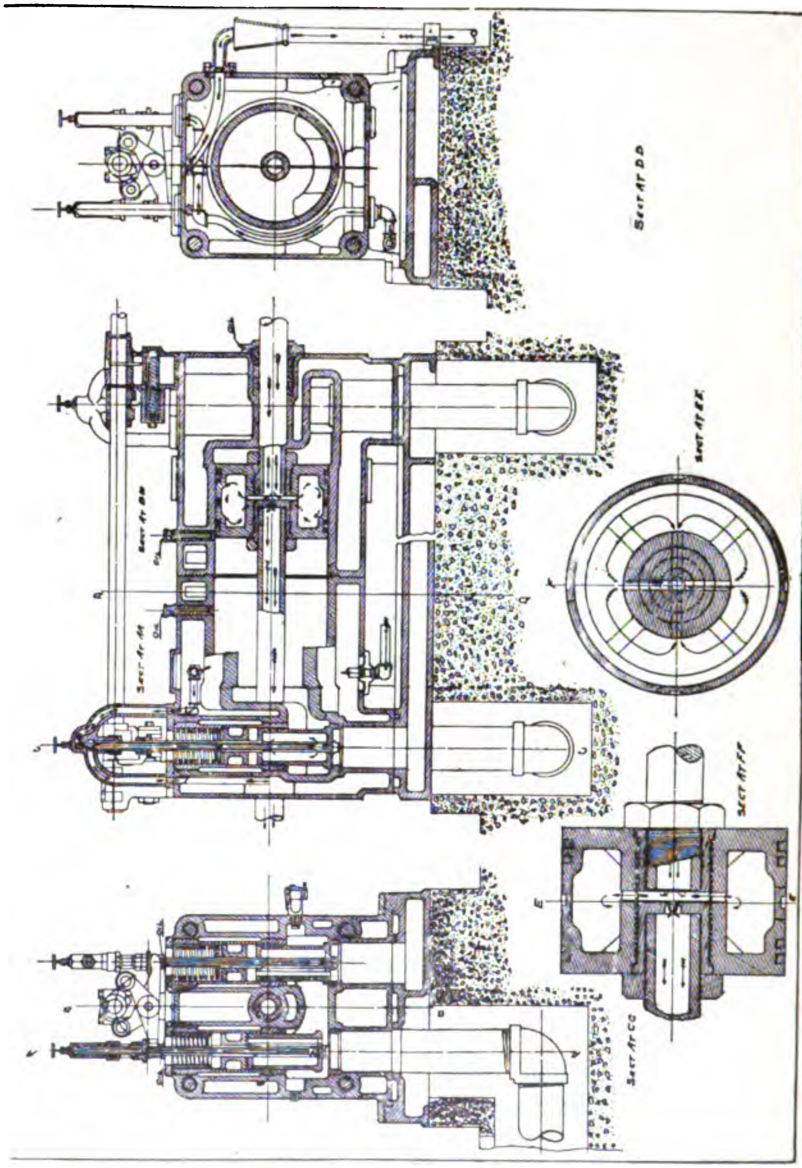


FIG. 12-65. — Details of Riverside Engine.

assuring positive alignment for these valves indefinitely. The valve seats are renewable and are located slightly below bottom of cylinder bore, so that all foreign substances are swept from the cylinder at each exhaust stroke. All valves are readily removed from the top of the cylinder without disturbing cam shaft.

Piston and piston rods are water-cooled, the water entering through a telescopic joint connected to side of cross-head. Circulation through each piston is positive and the overflow is so arranged that pistons are kept full of water. Water passages through piston and rod are large and easy so that not over ten pounds' pressure is required for circulation. The overflow is visible so that water cannot come to a boiling-point without attracting engineer's attention. A heavy tail-rod support and adjustable shoe is provided for carrying weight of pistons and rod. The construction of the piston rod is such that a piston in either cylinder can be removed without disturbing the other cylinder or the connecting rod, cross-head or any part of the valve gear.

The valve gear of the Riverside engine is very simple and consists of a single shaft mounted on top of the engine, running in self-oiling bearings. This shaft runs at one-half the speed of the main shaft and carries the inlet and exhaust cams, cams for operating the oil pumps, and the timers for ignition system: Power is transmitted to the inlet and exhaust valves by an inlet and exhaust lever hung on a single pin. All cams are keyed rigidly to the cam shaft. This construction makes a minimum number of joints subject to wear. Ample adjustments are provided for taking up wear.

Ignition is by an improved method throughout, consisting of two magnetically operated spark plugs in each cylinder. The sparking points being in series with the magnet coils give a positive external indication by the movement of the armature as to whether the spark takes place within the cylinder. The timing is electrical; no gearing or mechanical trips being used, simply a No. 14 stranded wire running through iron armored conduits leads to each plug.

The timer is mounted on the secondary shaft and is built in a heavy, substantial manner. The contacts are of the wipe type and are made of tool steel, and are adjustable for wear. A visible

indicating spindle shows the amount of contact. The entire wearing parts of timer are run in oil, which prevents burning of the contacts and reduces wear to minimum. Wiring from the timer to each spark plug is protected by a five ampere enclosed fuse; thus, should any spark plug become damaged or short-circuited, it would have no effect on the rest of plugs. Any spark plug can be removed and replaced while the engine is in operation.

The time of ignition in all cylinders can be adjusted simultaneously, and by an individual adjustment the time of ignition in any cylinder can be adjusted independently. All of these adjustments can be made when the engine is in operation.

The method of starting the double-acting engine is as follows:

A gas-engine driven air compressor is provided for supplying compressed air for starting this engine, air to be stored in a suitable steel receiver. The pneumatic starting gear on the engine consists of two shifter pistons mounted within the valve lever carrying pin. These pistons are shifted by throwing a small three-way cock which applies compressed air to one side of the piston, which in turn moves the valve levers $\frac{5}{8}$ inch, bringing the cam rollers in line with an auxiliary cam which puts all the valves in both ends of one cylinder into two-cycle action, or, in other words, makes a poppet valve air or steam engine out of this cylinder, which permits the engine being started on any stroke and on either quarter. The operation being entirely automatic, the engine will run as long as compressed air is applied. The other double-acting cylinder continues to operate as a four-cycle gas engine and takes up its explosions after the first revolution. Compressed air is then shut off from the starting cylinder and the three-way cock reversed, which permits the cam levers being returned to their normal position and that cylinder immediately goes into four-cycle action and takes up its explosions on the next revolution. The starting gear adds practically no complication to the engine as the regular inlet and exhaust valves are used for distributing the air. There are no extra shaft or auxiliary valves and no tappings into the cylinders. There is only one compressed air pipe leading to the sole plate, air being delivered into the fuel duct and entering the cylinders via the inlet valves.

The Crossley Engine. — This English engine is made in this

country by the Power and Mining Machinery Company of New York, and known as the American Crossley.

These engines are built as single-cylinder, two cylinder-opposed and double-opposed units, sizes ranging from 50 to 1300 B. H. P. This design differs radically from that of other large engines in that the single-acting cylinder is retained up to the largest sizes, *i.e.*, the design is in a way nothing but an enlargement of the small four-cycle machine. It is fundamently the same design employed at first by the Deutz Company for their large engines, but has been given up by them some years ago in favor of double-acting engines.

Figures 12–66 and 12–67 show a general elevation and a cross-sectional elevation respectively of the single-opposed type. The cylinder castings are securely bolted to each end of a central frame casting. Both connecting rods work on one crank pin. The pistons are water-cooled. The cylinder proper is a separate liner, resting at the crank end in the jacket casting and held at the head end by the cylinder head. The head carries the gas and inlet valves, while the exhaust valve is placed in a separate casting. The position of inlet and exhaust valve is shown in Fig. 12–68. Both are operated in a horizontal position by means of bell cranks by cams on a lay shaft. In this respect the engine differs from most others in which the valves are nearly always vertical. The objection to the horizontal form is at least partly overcome by giving the exhaust valve a guide at each end. This valve is water-cooled, as is necessary in large machines. The lay shaft *s*, Fig. 12–66, is operated by spiral gears, and actuates the main inlet, the cut-off valve, the gas valve, the exhaust valve, and the igniter gear.

Gas and air enter the mixing chamber *M*, Figs. 12–66 and 12–69 through a proportioning valve, the air direct and the gas through a special gas valve. From here the mixture passes a multi-port cut-off valve, which surrounds the valve stem of the inlet valve and is shown in greater detail in Fig. 12–70. This cut-off valve is operated from the lay shaft through the rocker arm *C*, Fig. 12–71. The governor, shown in plan in Fig. 12–69, serves to shift the fulcrum *f* of the rocker arm *c*, Fig. 12–71, thus cutting off the mixture supply to the main inlet valve earlier or later as the load demands. Should the load drop so far that the com-

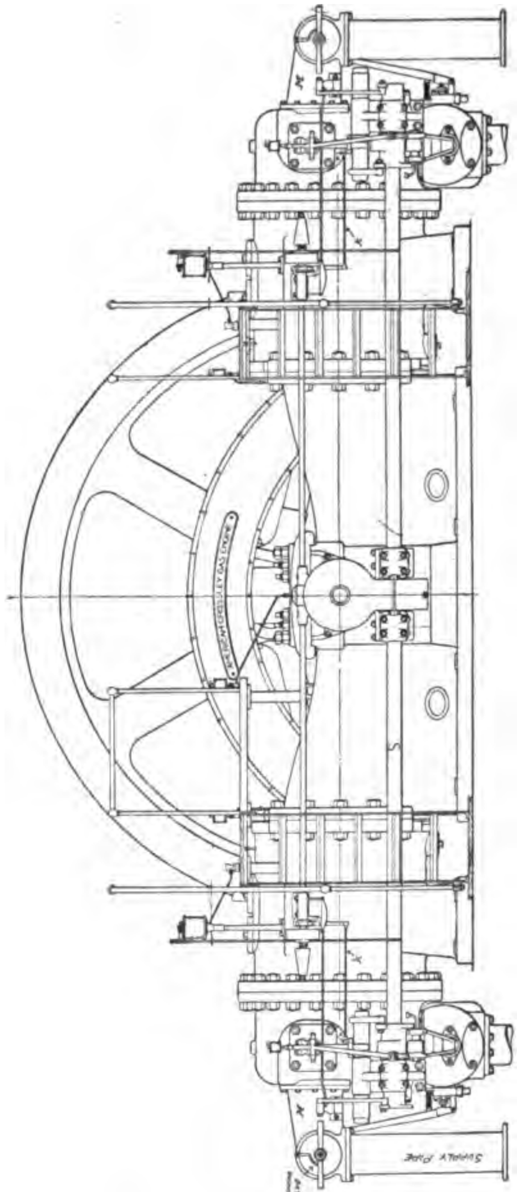


FIG. 12–66. — American Crossley Gas Engine.

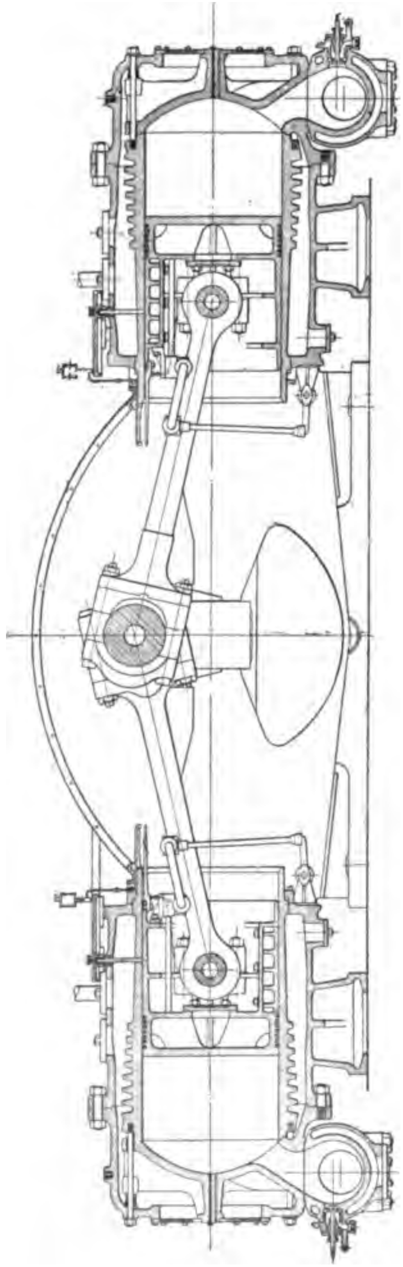


FIG. 12-67. — Cross-section American Crossley Gas Engine.

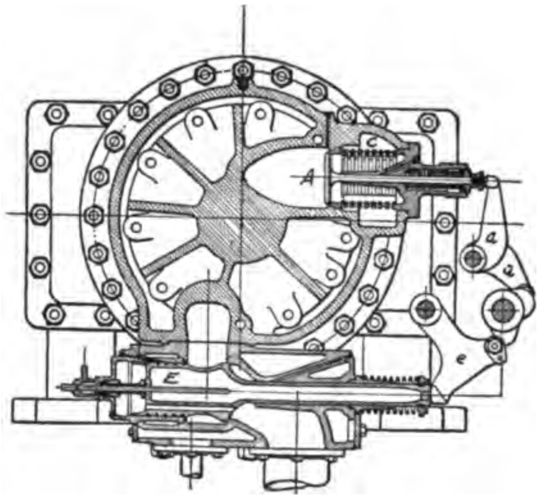


FIG. 12-68. — American Crossley Engine, Inlet Valve at A, Exhaust Valve at E.

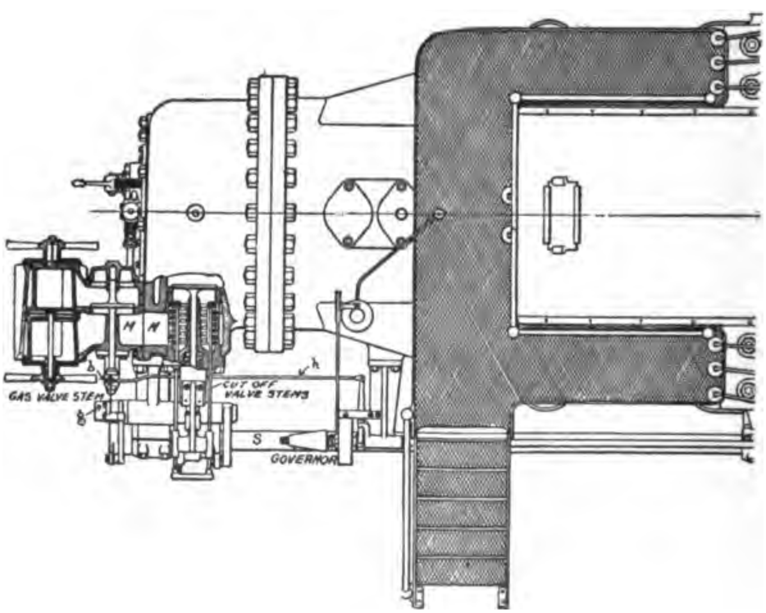


FIG. 12-69. — Details of Valve Gear, American Crossley Engine.

bustion becomes sluggish under very low loads, the engine automatically shifts over to the hit-and-miss governing by the governor withdrawing the block *b*, Fig. 12-69, thus keeping the gas valve closed altogether. In multi-cylinder engines the point at which this occurs is so adjusted that one cylinder after another changes over to the latter system of regulation and not all of them at the same time.

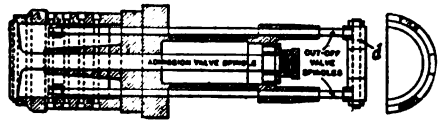


FIG. 12-70. — Inlet Valve, American Crossley Engine.

The American company for some time adhered to hot tube ignition, but some of the later engines were fitted with make-and-break electric ignition.

The Snow Engine. — The Snow Steam Pump Company build

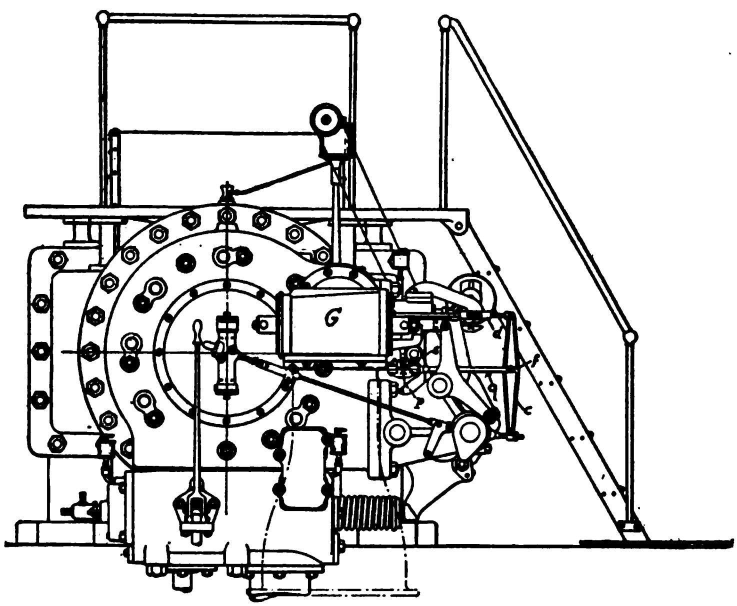


FIG. 12-71. — End View, American Crossley Engine.

two types of horizontal double-acting four-cycle machines, called respectively Type "A" and Type "B." Both of these designs differ radically from the construction of other large gas engines

in that the valves are placed in chambers at the side of the cylinders. There are reasons for and against this construction, the main disadvantage being perhaps the cut-up form of the combustion chamber. Some of the many advantages are outlined below.

Type B engines are built in single-cylinder units from 20 to 125 horse-power, and in tandem units from 80 to 500 horse-power. The general features of the design are shown in elevation in Fig. 12-72 and in cross-section in Fig. 12-73.

Fig. 12-72. — Snow Type B Gas Engine.

The most notable installations of Type A engines is found in the Martin station of the San Mateo Power Company in California. This station at present contains three of the twin-tandem units, a fourth is in course of erection, and the foundation is laid for a fifth. Fig. 12-74 shows a general view of the station, and also serves to give a general idea of the appearance of the engines. The engines of the Martin station were described in great detail by C. P. Poole in an article in Power, January 14 and

21, 1908, to which the writer is indebted for the following information:

Figure 12–75 shows in cross-section the main features of the design, while Fig. 12–76 is a transverse cross-section through one of the valve chambers to show the valve construction. Each cylinder of the twin tandem unit is 42 inches in diameter by 60-inch stroke. The engines are direct connected to Crocker-Wheeler generators rated at 4000 K.W. each. They are capable, however, of carrying momentarily an overload of 33 per cent, and have demonstrated their ability to carry 15 per cent overload continuously. These figures make these units the largest gas power

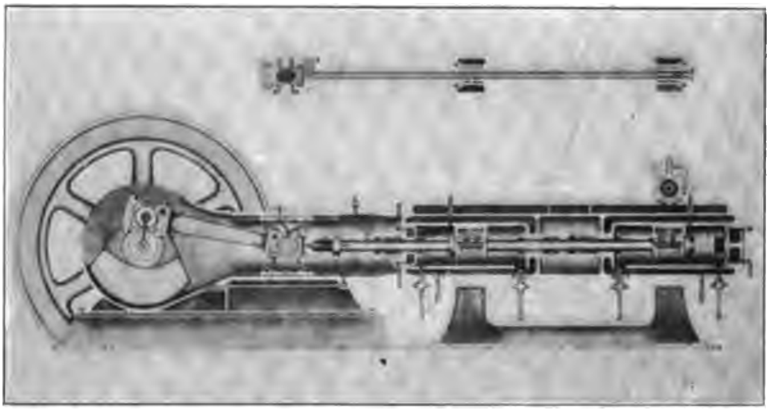


FIG. 12–73. — Cross-section of Snow Type B Gas Engine.

engines in the world. The fuel used is an oil-water gas made by the Lowe process.

The main frame is of the side crank type. The cylinders are cast in two parts, fastened together by flanges at the center, as shown in detail in Fig. 12–77. This construction makes the outer wall independent of the cylinder wall proper, and avoids temperature stresses. The valve chambers are cast integral with the cylinder casting. The cylinder heads are plain jacketed covers containing nothing but the stuffing-boxes. For the purpose of safety against leakage, however, each head has a double seat. The front and rear cylinders are connected by a distance piece of box-section, which has large openings in the side through which heads and pistons may be removed if required. The pistons are

FIG. 12-74. — Snow Type A Engines, Martin Station, San Mateo Power Co., Cal.

made in two parts bolted together. These pistons, together with their rods, are of course water-cooled, the water entering at the front cross-head and leaving at the tail cross-head.

The distinguishing feature of these engines, however, is the valve construction, shown in detail in Fig. 12-76. Mr. Poole, in the article mentioned, describes the gear as follows:

"The cam shafts which drive the valve gear are driven by steel gears, bevel gears being used on the main shaft, driving back through spur wheels to secure the proper reduction in speed of two to one. The igniters and lubricators are also driven from the cam shafts, as are also the starting valves, which admit compressed air on the cylinders for starting. The inlet and exhaust

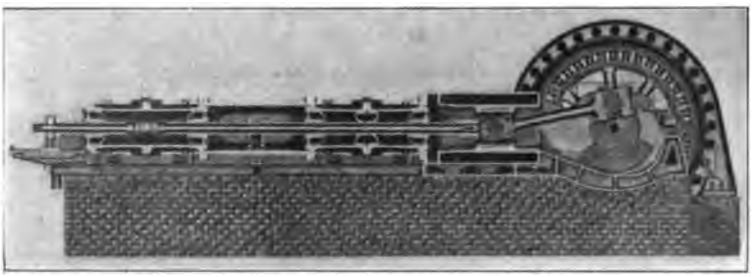


FIG. 12-75.

valves are driven by cams made of chilled cast iron, located on the cam shafts, which run in bearings bolted to the side of the valve chambers.

"The inlet and exhaust valves are of the unbalanced mushroom type, working in cages secured to the top and bottom of the valve chamber. Both valves and their cages are water-jacketed, in order to prevent back-firing or pre-ignitions on account of the treacherous nature of the gas used. Each inlet valve is combined with a combination mixing and throttling valve, of piston form, so designed that when the inlet valve opens gas and air ports in proper proportion are open for the passage of gas and air in the ratio desired, the amount of the opening of both being fixed by the governor. From this it will be evident that the engines operate with variable compression and constant mixture, the supply of air and gas to each end of each cylinder being throttled directly at each inlet valve for its end of its cylinder.

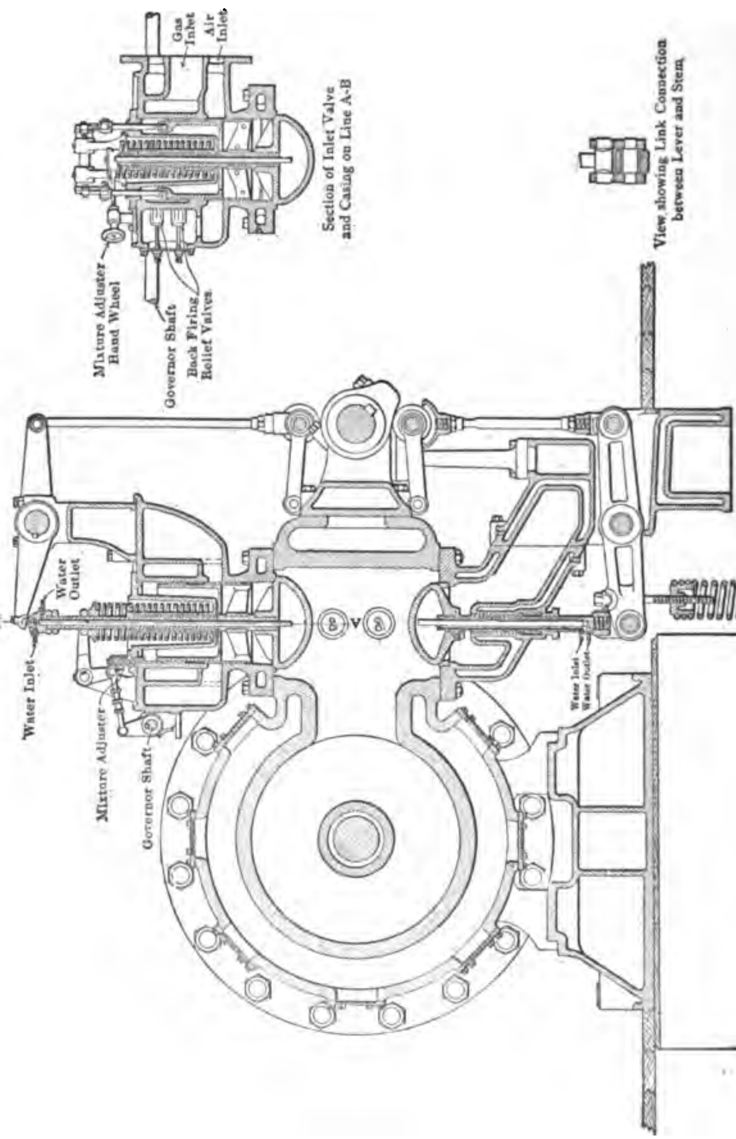


Fig. 12-76. — Transverse Section, Snow Type A Engine.

"The exhaust valves are of cast iron with nickel-steel stems and are thoroughly water-jacketed, the water being fed to and carried from the valve by positive circulation. The connection of the exhaust valve cages with the exhaust pipe is so made that the cages can be readily removed without disconnecting any other part of the engine."

In discussing the reasons for locating the valves at the side, Mr. Poole quotes the builders as follows:

"It permits of the use of an absolutely continuous bedplate under all cylinders, which is considered essential on engines as long as these for maintaining absolute alignment and permitting unrestricted movement to compensate for variations in temperature in cylinder walls and connecting parts.

Fig. 12-77. — Cylinder Construction, Snow Type A Gas Engine.

"A solid, unbroken foundation from one end of the engine to the other is thus secured, and the builders consider that a solid, unbroken foundation is more essential for a gas engine than for any other prime mover on account of the enormous weight of the reciprocating parts and the inability to fully counterbalance.

"It enables all working parts of the engine, without exception, to be located above the engine-room floor and therefore to be in full view of the attendants at all times.

"Inlet- and exhaust-valve driving motions from the cam shaft are short and direct.

"Exhaust valves are much more accessible for removal when located in a valve chamber on the side, since the crane can be used throughout the entire operation and all work in connection with the removal and displacement of the exhaust valves is done from the engine-room floor.

"With valves located in a side valve chamber, broken inlet and exhaust valves cannot get into the interior of the cylinder and lodge between the piston and the cylinder-head, causing wrecks.

"It has been very generally the opinion of gas-engine designers that the location of the valves in a side valve chamber of the kind used on the California engines entailed the certain disadvantage that foreign material entering the cylinder with the air and gas would deposit on the bottom of the cylinder counterbore, while if the exhaust valve be located on the bottom, such deposits will be carried out through that valve; furthermore, that lubricating oil collects in the bottom of the counterbore and is carbonized, causing back-firing and pre-ignitions. Experience with these engines thus far indicates that if lubrication is properly effected no carbonized oil is found in the cylinder, and that when oil is supplied excessively the resulting deposit of carbon is located not on the surface of the clearance space, but on the top of the piston barrel directly under the oil-inlet holes, on the piston rod where it wipes in from metallic packing, and on the face of the cylinder head directly under and close to the piston-rod hole in the head."

The Cockerill Engine. — This machine is built by the Soc. Anon. John Cockerill in Seraing, Belgium. This firm builds several types of engines, differing among themselves mainly in their valve constructions. Fig. 12–78 * shows a 1200 horsepower tandem gas engine, in which the inlet valve is on top, the exhaust valve on the bottom of the cylinder. Both are operated apparently from the same cam on the lay shaft. This type of machine is built either for quality or quantity governing, in which case the construction of the inlet valve differs. Fig. 12–79 shows another form, a single-cylinder double-acting engine direct connected to a blowing cylinder. The details of the valve gear of this machine are quite clearly shown in Fig. 12–80.† Both inlet and outlet valves are side by side at the bottom of the cylinder.

Figure 12–81 shows the latest form of the double-acting tandem Cockerill engine.‡ The distinctive features of this machine are that all valves are placed below the cylinder and that eccentrics only are used for operating these valves. The valve gear proper represents one of the most complicated constructions used

* H. Dubbel, Z. d. V. d. I., Sept., 1905.
† Güldner, p. 479.
‡ Güldner, p. 481.

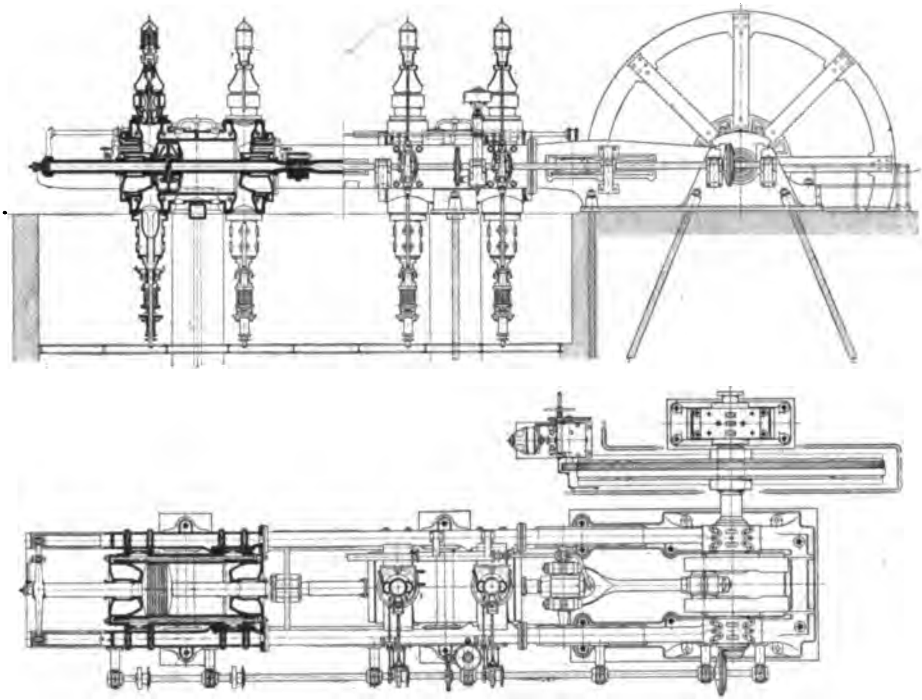


FIG. 12-78.

FIG. 12-79.

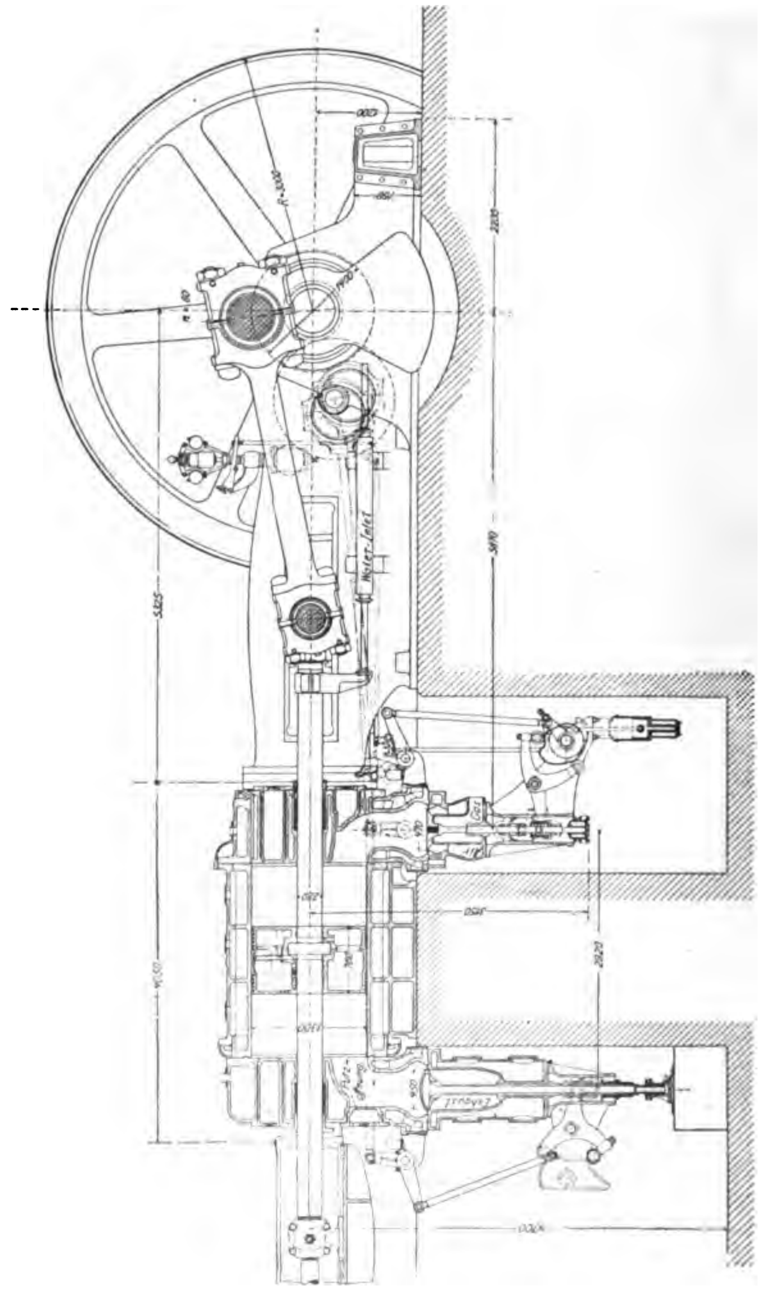


FIG. 12–80.

to-day and is shown in section in Fig. 12-82. In this figure at the left, the eccentric, through its rod *a* and the lever *b*, operates the cam *c* which positively opens and closes the exhaust valve through the triple lever *d*. The inlet valve eccentric, Fig. 12-82, at the right, in a similar manner operates the gas valve cam *c* and the inlet cam *c′*. The lever *b* actuating the gas valve cam is disengaged from the rod *a* when the trip arrangement *f–f′* is brought into play by the cam disc *e*. The position of this disc on its shaft is regulated by the governor, through the linkage shown, thus effecting speed regulation. The dash pot *h* allows the gas valve to seat noiselessly after its linkage is released.

FIG. 12-81. — Cockerill Double-acting Tandem Engine

The Cockerill engine has found considerable application in Europe, there being about 80,000 B. H. P. in operation in 121 engines, ranging from 200 to 3000 B. H. P. The fuel used is mainly blast furnace gas.

The Nürnberg Engine.—Although the Allis-Chalmers Company of Milwaukee have been the American licensees of the Maschinen-Gesellschaft Nürnberg, makers of the Nürnberg engine, the large gas engine now turned out by the American firm, differs in some of the important details from the original German design.

The Nürnberg engine is to-day perhaps the most important four-cycle gas engine built in Germany, and for that reason a description is first given of the German machine, to be followed by a few details of the Allis-Chalmers engine as far as they could be obtained.

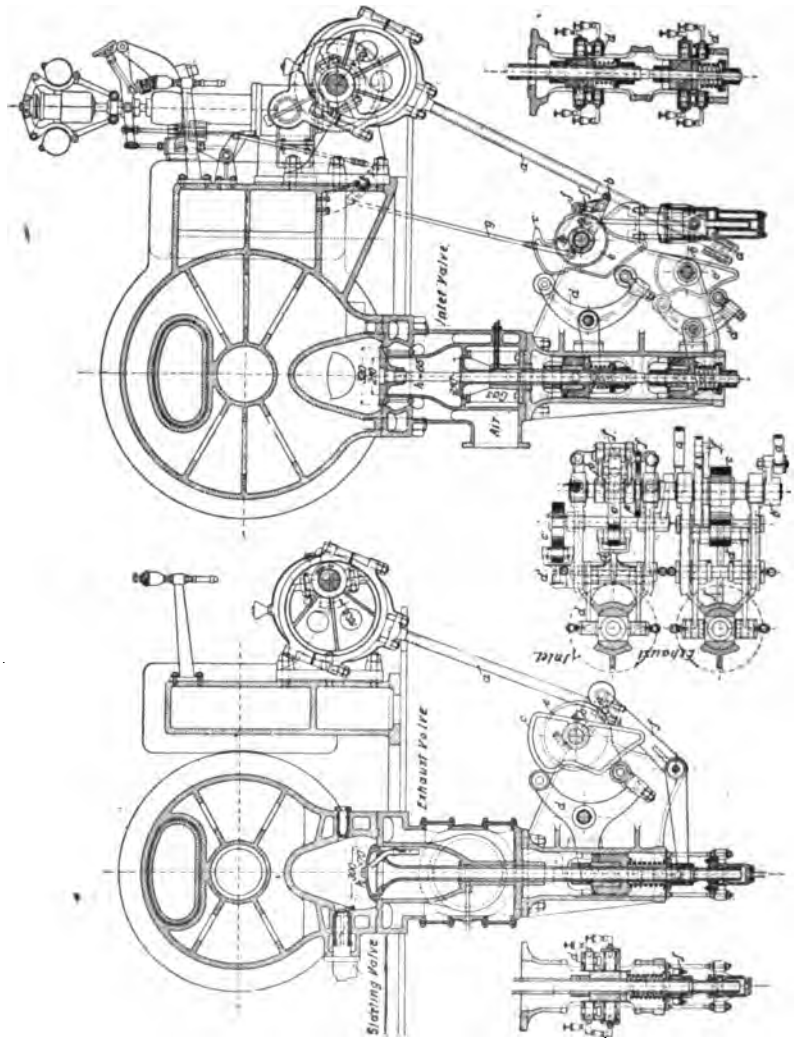


Fig. 12-82. — Valve Gear Details of Cockerill Double-acting Tandem Engine.

All of the medium sized and large Nürnberg engines are double-acting, either single-cylinder or tandem. They range in size from 250 to 6000 horse-power, the larger powers being the twin-tandem type. Probably the best description ever given of this machine is contained in Riedler's "Gross Gasmaschinen," from which the cross-section, Fig. 12–83, together with the two following figures, is taken. This cut shows frame, cylinder, and valve construction very clearly. The only part rigidly fastened to the foundation is the main frame, which is of the center crank type, as preferred by all European builders. The cylinders rest on supports such that they can freely expand and contract under temperature changes. The gas and inlet valves are placed on top and the exhaust valve at the bottom of the cylinder. The inlet and exhaust ports are cast in one part with the cylinder and jacket walls. This reduces the cylinder heads to simple water-jacketed covers, a point of great advantage as compared with the complicated cylinder heads of former days. All valve cages with their valves are easily removable for inspection. The exhaust valves are water-cooled. The valves are operated from a lay shaft running alongside, Fig. 12–84, by means of eccentrics and not by cams. How this motion is transmitted to the valve stems by roller levers to avoid shock and noise is very clearly shown in Fig. 12–85. The pistons are as simple as those of a steam engine. The method of fastening them on the shaft together with the water-cooling arrangements for rods and pistons are indicated in Fig. 18–83. The weight of piston and rod is carried by outside bearings, which relieves both cylinder and stuffing-boxes.

The greatest attention to accessibility has been paid in the design of this engine. As shown in Fig. 12–86, by disconnecting the piston rod between the cylinders, disconnecting the connecting rod and sliding the rod and cross-head forward, and by taking off the front and rear cylinder covers, the entire engine interior is at once open to inspection.

The Nürnberg engine regulates by pure quality regulation, that is, the amount of gas only is cut down as the load drops. To accomplish this there is a gas valve placed ahead of each main inlet valve on the top of the cylinder. Fig. 12–83 shows the construction of these valves in section and the method of operating them by eccentrics from the lay shaft may be seen in Fig. 12–84.

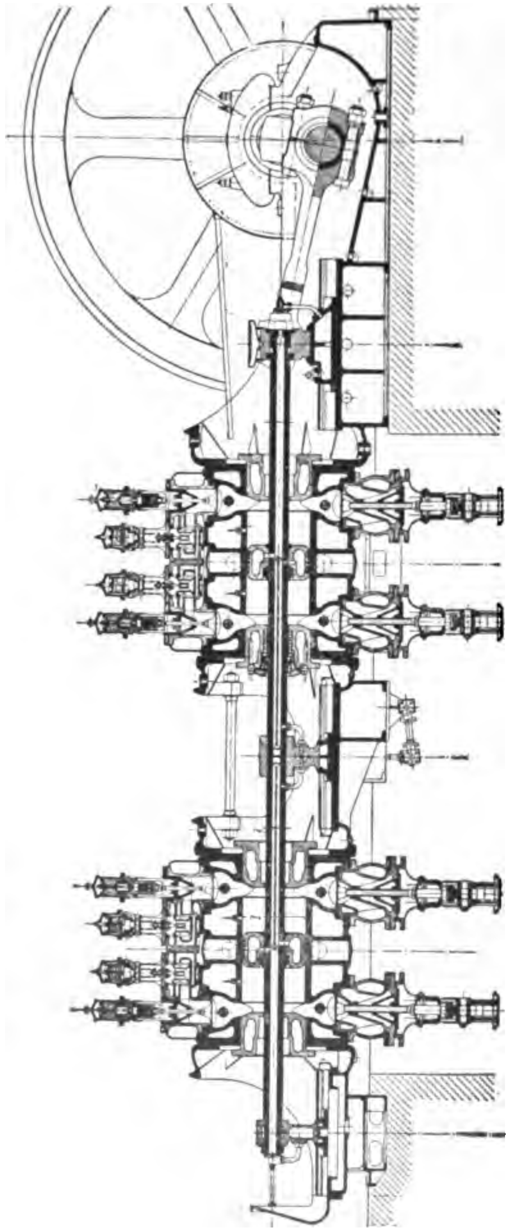


FIG. 12–83. — Nürnberg Double-acting Tandem Gas Engine.

FIG. 12-84. — Nürnberg Double-acting Tandem Engine.

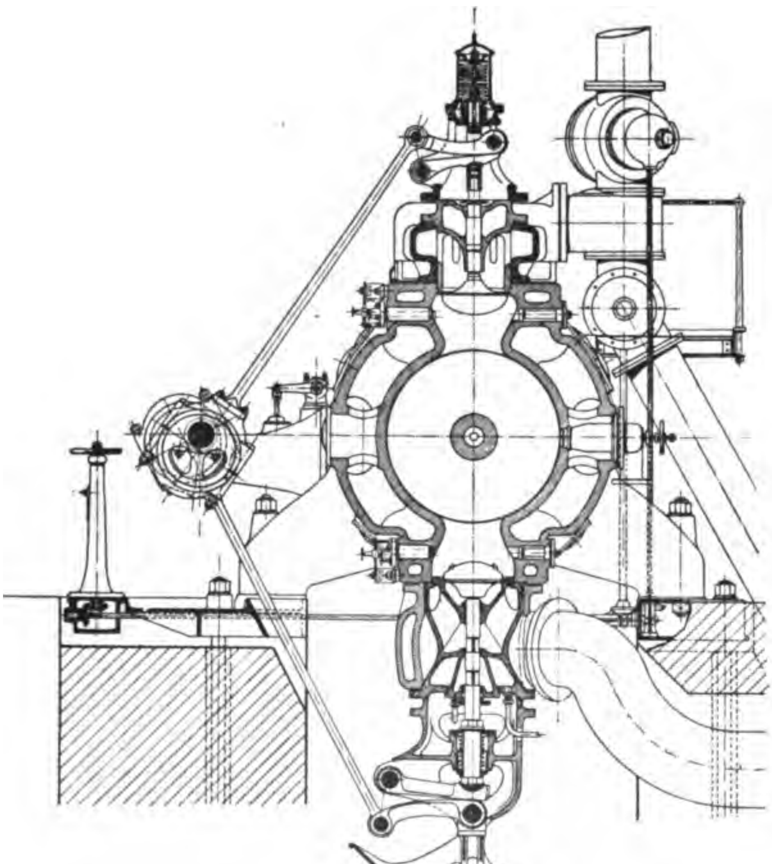


FIG. 12-85. — Details of Valve Gear, Nürnberg Engine.

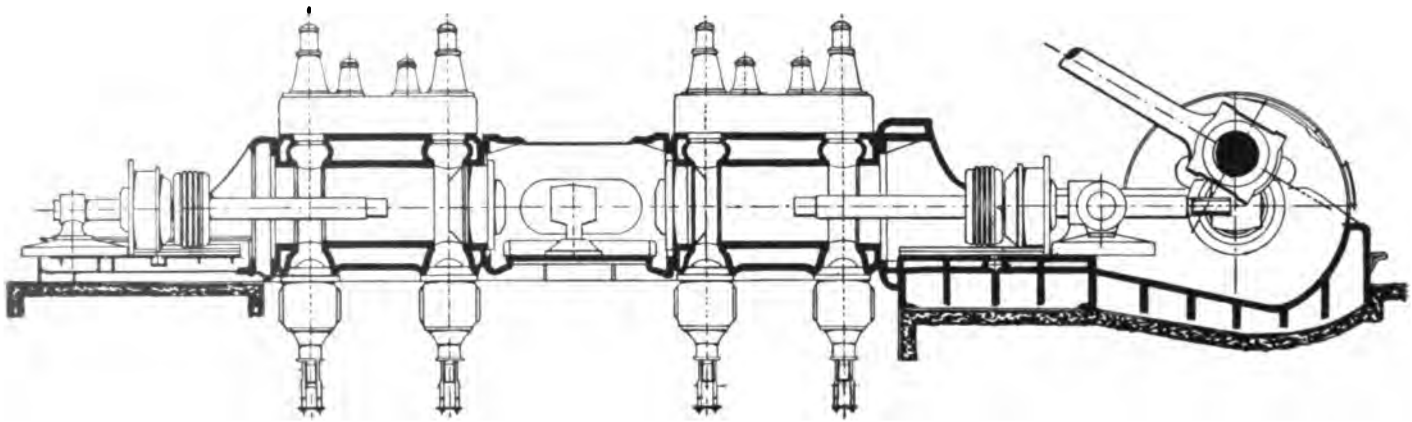
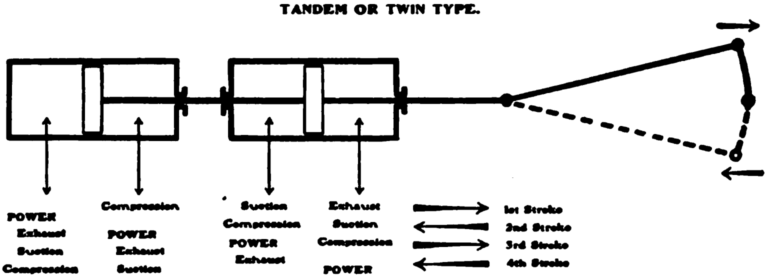


Fig. 12-86. — Showing Accessibility of Nürnberg Engine.

The latter illustration also shows the centrifugal governor and the governor shaft running alongside of the cylinders. From this governor shaft reach rods run to the gas valves and control the time of the opening of these valves in the manner explained in Chapter XIV. In some of the later engines the details of the governor mechanism are changed somewhat * but the principle of operation is the same in all, *i.e.*, to open the gas valve later in the stroke as the load drops and to keep the time of closure the same. The air is not throttled at any time, hence the compression remains about the same. It is interesting to trace out the sequence of events in a four-cycle tandem engine, for which purpose Fig. 12-87 published by the Allis-Chalmers Company is given.

SEQUENCE OF OPERATIONS IN
THE FOUR-CYCLE, DOUBLE-ACTING SYSTEM.
TANDEM OR TWIN TYPE.



4 POWER STROKES IN 2 REVOLUTIONS.

FIG. 12-87.

Most Nürnberg engines operate on blast furnace gas. The table on page 346, also from the catalogue of the Allis-Chalmers Company, will serve to give some idea of the cylinder sizes, speeds, floor-space, etc.

It has already been mentioned that the engine now built by the Allis-Chalmers Company is not quite like the Nürnberg machine. The following is a description of one of these engines, as given in Power, January 21, 1908. The engine, Fig. 12-88 is of the twin-tandem, double-acting type and direct connected to a 1040 K.W. Crocker-Wheeler generator:

"It is of the side-crank type so generally used in steam engine

* See Zeitschrift des Vereins deutscher Ingenieure, Aug. 11, 1906, and Sept. 22, 1906.

construction. The pistons are supported entirely by the piston rods, which are turned with sufficient camber to make them dead straight when the weight of the pistons is put on them. The rods are equipped with intermediate and tail shoes, as usual with large tandem construction, running on guides between the cylinders and behind the rear cylinder, in addition to the main cross-head.

NÜRNBERG GAS ENGINE

TABLE OF STANDARD SIZES

Tandem						Twin						Twin-Tandem					
Size of Cylinders				Floor Space		Size of Cylinders				Floor Space		Size of Cylinders				Floor Space	
Diam. Inches	Stroke. Inches	Speed. Revolutions per Min.	Rated Power. B. H. P.	Length Rear End to c. of Shaft. Feet	Width Overall. Feet	Diam. Inches	Stroke. Inches	Speed. Revolutions per Min.	Rated Power. B. H. P.	Length Rear End to c. of Shaft. Feet	Width Overall. Feet	Diam. Inches	Stroke. Inches	Speed. Revolutions per Min.	Rated Power. B. H. P.	Length Rear End to c. of Shaft. Feet	Width Overall. Feet
18	24	150	260	29	11	18	24	150	245	19	17	18	24	150	520	29	17
20	24	150	320	29	11	20	24	150	305	19	17	20	24	150	640	29	17
21	30	125	370	33	13	21	30	125	350	22	20	21	30	125	740	33	20
22	30	125	405	33	13	22	30	125	385	22	20	22	30	125	810	33	20
24	30	125	490	33	13	24	30	125	465	22	20	24	30	125	980	33	20
24	36	115	545	38	15	24	36	115	515	25	23	24	36	115	1090	38	23
26	36	115	630	38	15	26	36	115	600	25	23	26	36	115	1260	38	23
28	36	115	740	38	15	28	36	115	700	25	23	28	36	115	1480	38	23
30	42	100	855	44	17½	30	42	100	810	29	27	30	42	100	1710	44	27
32	42	100	985	44	17½	32	42	100	935	29	27	32	42	100	1970	44	27
34	42	100	1105	44	17½	34	42	100	1050	29	27	34	42	100	2210	44	27
36	48	92	1300	50	21	36	48	92	1240	33	32	36	48	92	2600	50	32
38	48	92	1460	50	21	38	48	92	1400	33	32	38	48	92	2920	50	32
40	48	92	1630	50	21	40	48	92	1550	33	32	40	48	92	3260	50	32
42	54	86	1875	60	26	42	54	86	1780	40	38	42	54	86	3750	60	38
44	54	86	2080	60	26	44	54	86	1980	40	38	44	54	86	4160	60	38
46	54	86	2280	60	26	46	54	86	2170	40	38	46	54	86	4560	60	38
48	60	78	2475	70	30	48	60	78	2350	45	44	48	60	78	4950	70	44
50	60	78	2720	70	30	50	60	78	2580	45	44	50	60	78	5440	70	44
52	60	78	2950	70	30	52	60	78	2800	45	44	52	60	78	5900	70	44

NOTE — The overall widths given in the above tables allow for a plain fly-wheel: proper addition must be made if a band-wheel is substituted or an electric generator direct connected upon the crank-shaft.

"There are two inlet valves to each combustion chamber, one for air and one for gas, but both are mounted in a single cage. The air valve is of the piston type and the gas valve is of the poppet type, its axis being in line with that of the air valve. The latter is opened at the beginning of the suction stroke and

kept open throughout that stroke. The gas valve, however, is open during varying proportions of the suction stroke, according to the requirements of the load. The engine, therefore, operates with constant compression and varying mixture proportions. The compression pressure is about 180 pounds per square inch, absolute.

"The ignition is by make-and-break mechanism electrically operated; that is, the movable electrode is rocked by means of an electromagnet and the latter is energized by the ignition current controlled by a 'timer' or so-called commutator driven synchro-

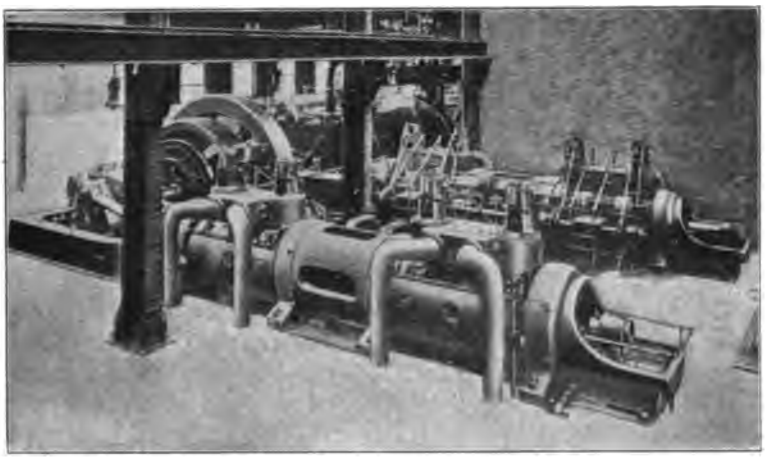


FIG. 12–88. — Allis-Chalmers Double-acting Twin Tandem Engine.

nously with the cam shaft. Tell-tale incandescent lamps are connected in circuit with the igniter magnets and indicate the performance or default."

The main differences between this and the Nürnberg engine then appear to be different cylinder supports, the combination of the gas and inlet valves in one housing, and the use of the side-crank frame.

Premier Engine. — Probably the best description of this engine is found in Robinson's "Gas and Petroleum Engines." It is of what is known as the positive scavenger type, *i.e.*, the combustion chamber is thoroughly washed out with air before a new charge is admitted. The cylinder is single-acting, and single-

cylinder and tandem units are built. The former go up to 250 horse-power, the latter up to 1200 horse-power. The pump serving to furnish the air for scavenging is usually placed ahead of the first power cylinder, but in some of the later designs the air pump is placed obliquely above the first cylinder.

Figures 12–89 and 12-90 are two illustrations from Robinson. The first shows the general appearance of the machine, the method of operating the inlet and exhaust valves from the lay shaft, the oblique rod to operate the air-pump valve, and the governor details. The second shows several sectional views from which the operation may be explained. The piston *D* of the single-acting air pump is rigidly connected to the piston *C* of the first power cylinder. The second power piston is connected by means of the yoke *K* and the side rods *R R* to the air pump piston. Thus the use of a stuffing-box in the combustion chamber of the forward power piston is avoided. *H H* are the combination air and gas inlet valves, and *G G* the exhaust valves, operated as shown in the transverse cross-section.

The operation is as follows: On the out stroke, air is admitted to the air pump *D* through the grid valve *F*, shown in the section *X X*. On the return stroke this air is compressed into the passage *D–E*, and serves to scavenge out the power cylinder just exhausting when the inlet valve *H* into that cylinder is opened about one-half exhaust stroke. On the next out stroke the air pump takes a new charge, but at the same time the valve *F* has also opened communication to the passage *E*, so that the cylinder just charging may draw air freely. Above the inlet valve *H* there is placed a cylindrical valve, which during the scavenging action opens the air ports completely and keeps the gas ports closed, but which, at the beginning of the charging stroke, shifts its position to give the proper relation between air and gas. While the exhaust and charging strokes above described take place in one, say the first, cylinder, the second has completed its compression and expansion strokes. On the third (the in stroke) of the pump, therefore, we shall have compression in the first cylinder and exhaust in the second. The pump then scavenges out the second cylinder during the last half of its exhaust stroke.

Speed regulation in this engine is effected by the governor cutting out the gas from the back cylinder altogether below a

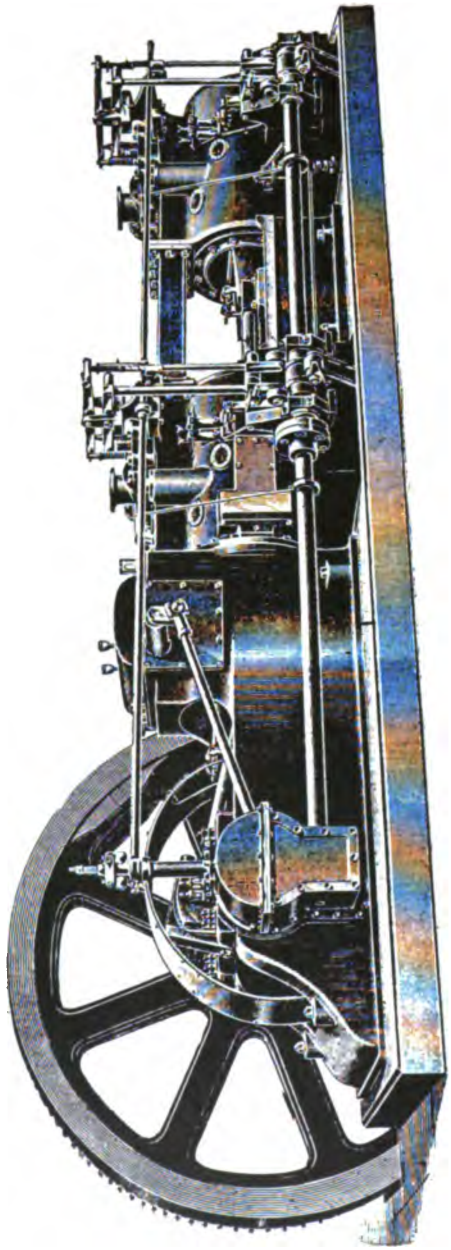


FIG. 12-89. — Premier Gas Engine.

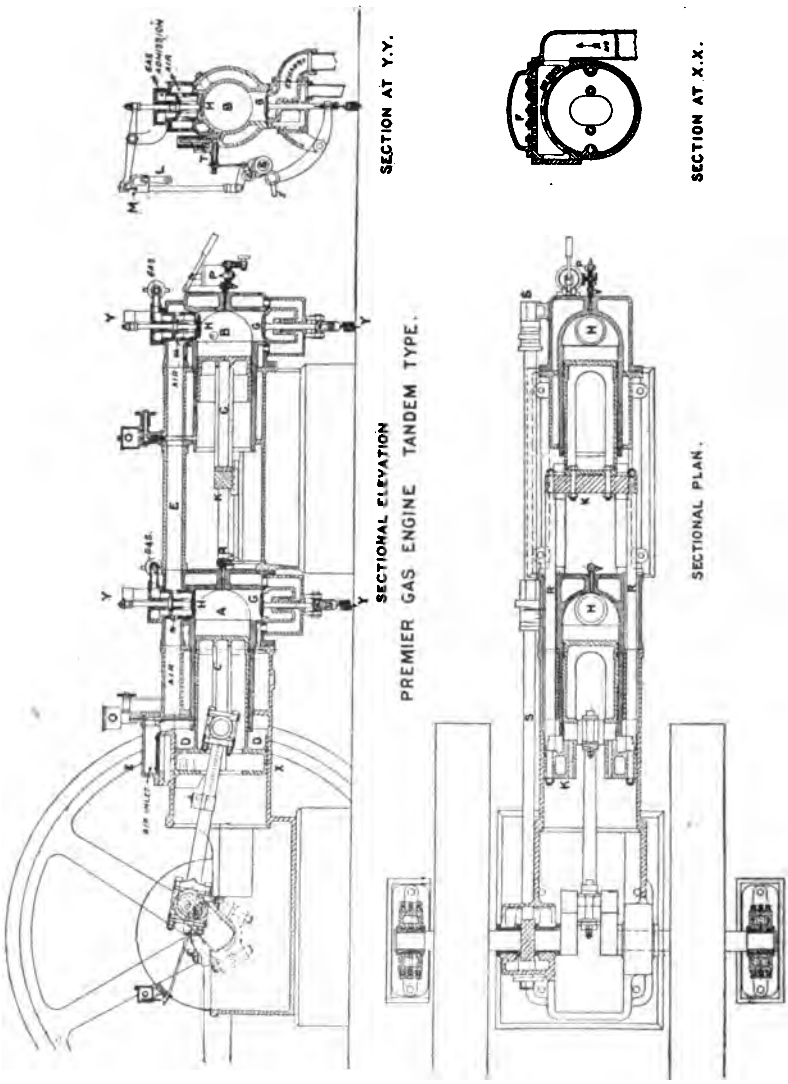


FIG. 12-90. — Details of Premier Gas Engine.

certain load. Above this a charge of gas is admitted occasionally, while above half power both cylinders work practically continuously.

Robinson states that the tandem design and the positive scavenging action make the Premier engine well adapted for the burning of cheap fuel gas. Güldner judges, from the high efficiencies obtained in conjunction with mean effective pressures exceeding 110 pounds per square inch, that the combustion process in this engine must be very perfect.

The Deutz Engine. — The Deutz Company of Deutz-Cologne were the first firm to build Otto engines and they have from the first been prominently identified with the development of this type of engine. Their present-day constructions embrace several designs of excellent form.

The large engines are all double-acting and are made either single-cylinder, twin or tandem. The general features of the design of a 600 B. H. P. twin engine are shown in longitudinal cross-section in Fig. 12–91 and in transverse section in Fig. 12–92.* The cylinder differs from that of the Nürnberg engine in that the jacket wall is not continuous. This is done, of course, to prevent temperature stresses. The central space is closed by a saddle casting as shown. Inlet and exhaust valves are placed in separate cages easily removable. The details of piston design and the water-cooling arrangements are clearly indicated. The manner of operating the valves from the lay shaft is shown in Fig. 12–92. All Deutz engines govern by throttling a mixture of constant composition. In the particular engine under discussion the throttle valve is a multi-ported cylinder surrounding the inlet valve stem. It is so designed as to keep the ratio of gas to air the same as that set by the hand-operated valves shown in the gas and air passages, but depending on the load, the mixture is throttled more or less. How the movement of the valve is controlled by varying the position of the fulcrum about which the throttle valve lever turns can be easily traced out by following the governor linkage.

The arrangement of the throttle valve is somewhat different in smaller engines. Thus in the 250 B. H. P. engine, a cross-section of which is shown in Fig. 12–93, the throttle valve is

* From H. Dubbel, Zeitschrift d. V. d. I., Sept. 2, 1905.

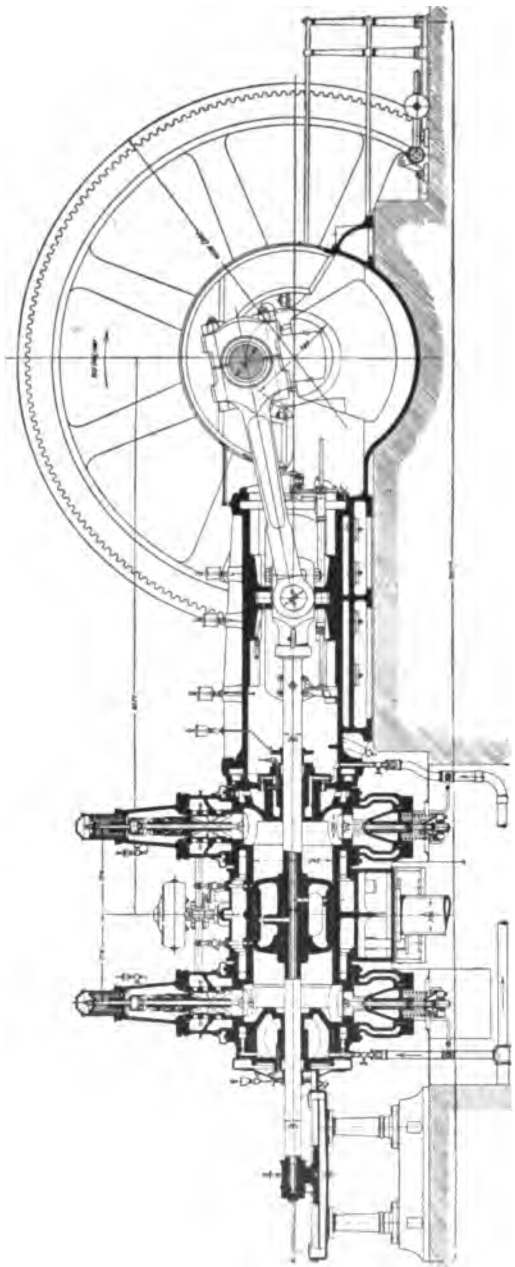


FIG. 12-91. — 600 B. H. P. Deutz Engine.

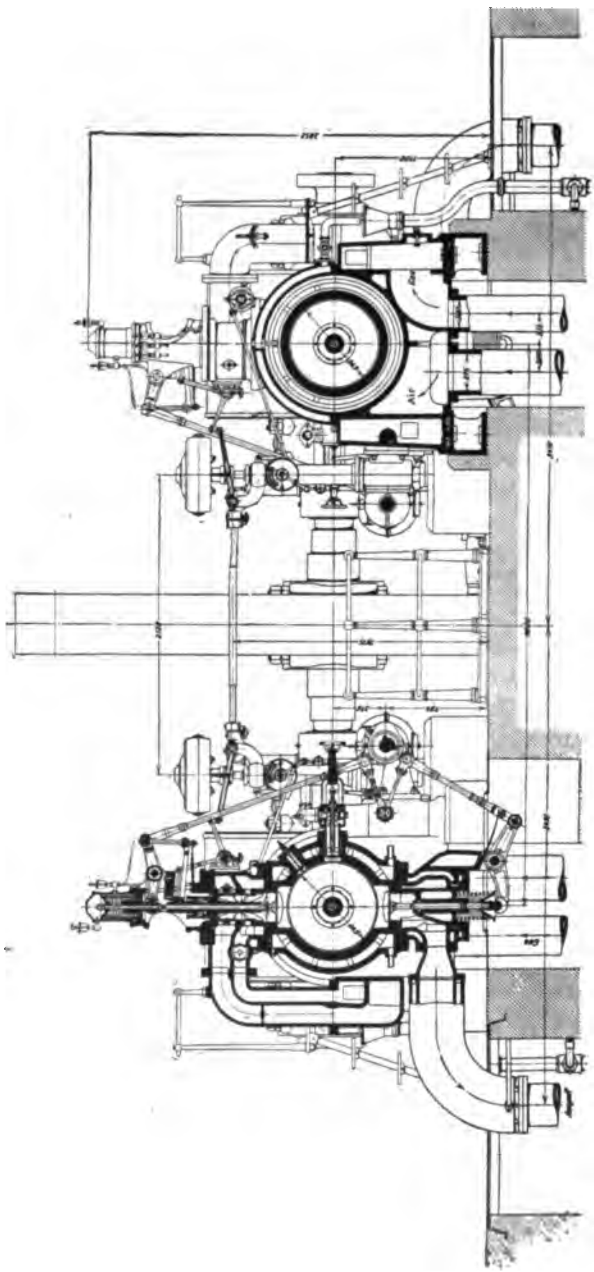


FIG. 12-92. — 600 B. H. P. Deutz Engine.

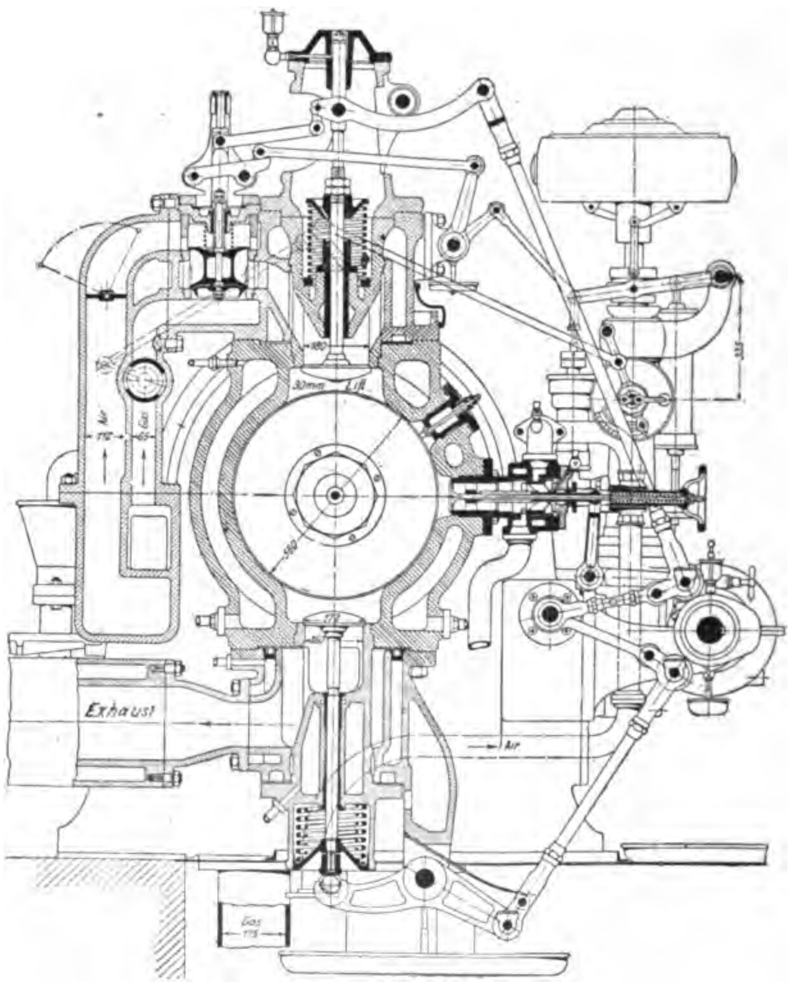


FIG. 12-93. — Valve Gear Details, 250 B. H. P. Deutz Engine.

independent of the main inlet valve. The governing features, however, are quite similar.

The Oechelhäuser Engine. — The Oechelhäuser Engine, built by several firms in Germany, is a two-cycle machine of distinctive design, as may be seen from the cross-sections, Figs. 12–94, which represents a 1000 horse-power machine built by A. Borsig.*

The power cylinder is open at both ends and contains two single-acting pistons. The front piston is connected to the middle crank of a three-throw crank shaft by piston and connecting rod in the ordinary manner. The rod of the back piston is fastened to a yoke which transmits the motion through rods on each side of the cylinder to cross-heads and connecting rods working on the two outside crank pins. This construction, while complex and probably costly, results in almost perfect balance. There are no valves as such. The exhaust gases escape through a ring of ports which is uncovered by the front piston near the end of the out stroke. Similarly the back piston near the end of its out stroke uncovers first a ring of ports admitting a charge of air for scavenging, and a moment later the gas or mixture ports. On the return stroke these rings of ports are covered one after the other and compression ensues between the two pistons. When in their inner dead center position, ignition is made to take place electrically, and the pistons are driven apart on their power strokes.

Gas and air under certain small pressures are furnished to the receivers surrounding the inlet ports by means of a combination air and gas pump which is usually operated by a tail rod. In blowing engines the blowing cylinder is usually so connected, and the air and gas pump is then placed under floor.

At full load the mixture follows the preliminary charge of pure air used for scavenging to about three-quarters or seven-eighths of the stroke. Regulation is effected by controlling the amount of mixture admitted to the cylinder. The means for doing this are discussed in Chapter XIV.

Oechelhäuser engines have given good satisfaction on blast-furnace and coke-oven gas. The general appearance of a twin engine of this type is shown in Fig. 12-95.

* Hoffman, Zeitschrift d. V. d. I., Sept. 15, 1906.

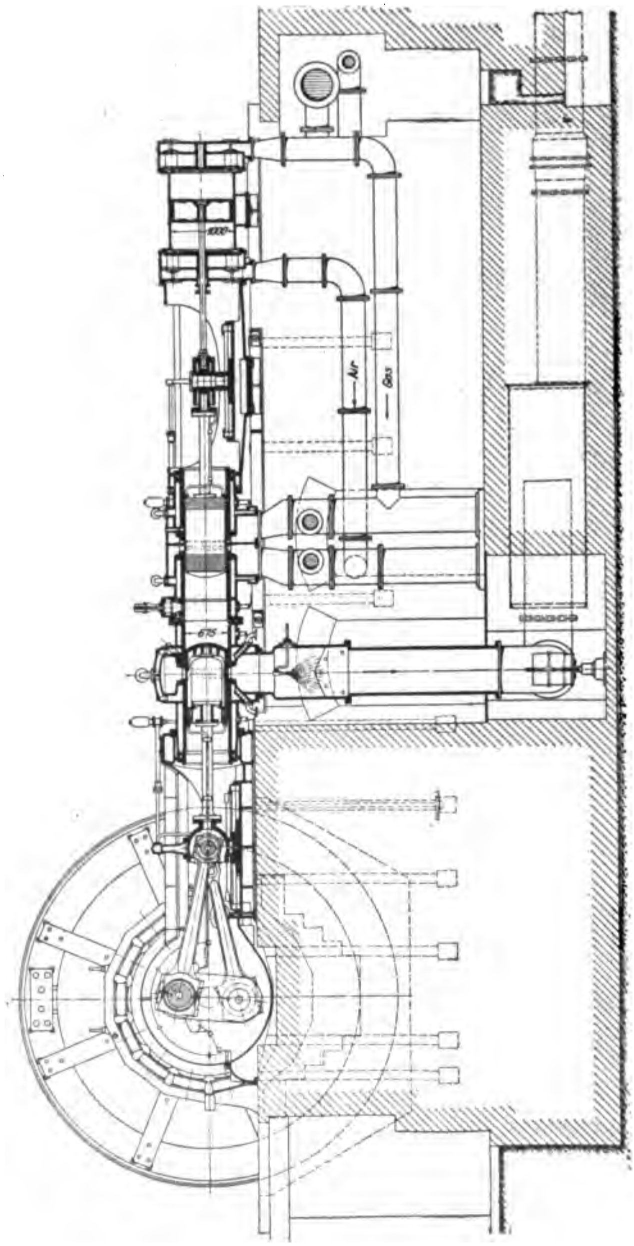


FIG. 12–94. — 1000 H. P. Borsig-Oechelhäuser Two-cycle Engine.

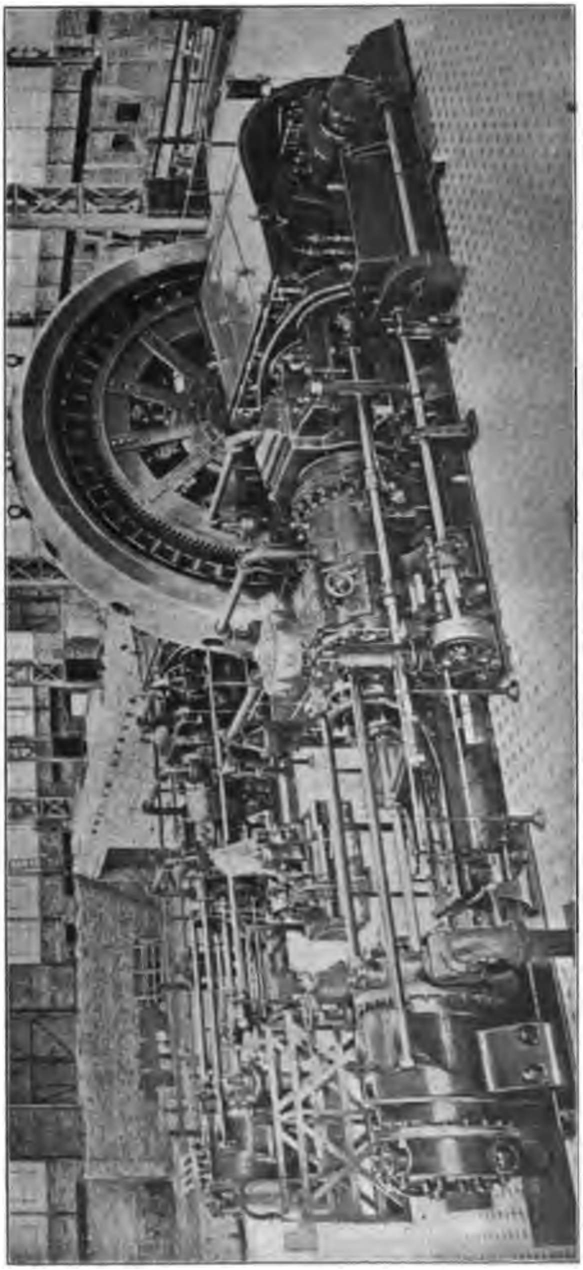


FIG. 12-95. — Twin Type Oechelhäuser Two-cycle Engines.

B. Liquid Fuel Engines

3. Gasoline Engines. — The very great majority of all portable internal combustion engines are gasoline engines, and the same is true of a large percentage of stationary internal combustion engines for small powers. We may thus divide all gasoline engines into three classes — stationary engines, marine engines, and automobile engines. The number of firms engaged in the manufacture of these engines is very large. Usually the various makes differ only in minor detail, and for that reason only one or two *type examples* under each class will be considered.

Stationary Gasoline Engines. — The general type of this machine either vertical or horizontal does not differ much from that of the small-powered stationary gas engine, several makes of which have already been discussed. It is usually possible to convert any gas engine of this type into a gasoline engine by the simple addition of a carbureting device. A number of the latter have been described in Chapter VIII.

The Foos Gasoline Engine. — This machine is made by the Foos Gas Engine Company of Springfield, Ohio. The cylinder, Fig. 12-96, is partly supported by the frame, making an especially rigid construction. The cylinder head is a plain flat cover. Both valves, which are of the poppet type, work upward and are mechanically operated; they are not affected by any governor action. Thus if the governor keeps the fuel valve closed, pure air is worked through the engine, cooling the cylinder and clearing it thoroughly from burned gases. The valves are operated by bell cranks and rods as shown. These rods are in turn actuated by cams on the shaft of the secondary gear, *A*, Fig. 12-96. The valves have their seat castings separate from the cylinder, but in order to remove the valves for inspection or regrinding it is necessary only to remove the plugs in the top of the valve cages. In Fig. 12-96, *B* is the rod operating the fuel valve, *C* the fuel pump, and *D* the igniter rod. The fuel may be gasoline, naphtha, or distillate.

Foos engines are governed on the hit-and-miss principle. In Fig. 12-97, *E* is the igniter rod, *C* the fuel valve lever which is oscillated by a cam on the shaft of the secondary gear *G*. This gear, through a pinion, also drives the governor *D*. The latter is

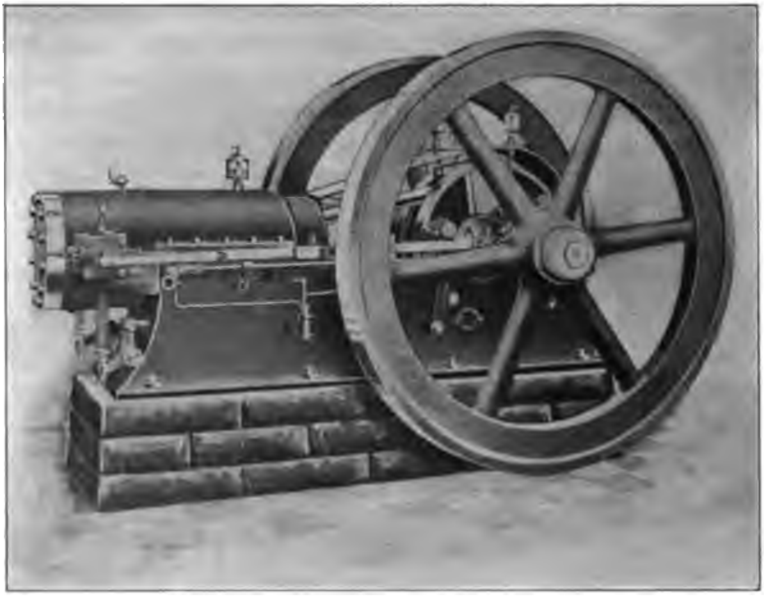


FIG. 12-96. — Foos Gasoline Engine.

FIG. 12-97. — Governing Details, Foos Engine.

of the fly-ball type, and, when the speed becomes excessive, pulls the block *B*, on the lower end of the governor lever *A*, out of line with the blade *F*. Thus *F* misses the fuel valve rod and the engine fails to receive a charge. The speed of the engine may be adjusted during operation by varying the position of the fulcrum *A* by turning the nut *H*.

Fig. 12-98. — Wipe Spark Igniter, Foos Engine.

In place of the ordinary hammer break system, a wipe spark igniter is employed. In Fig. 12-98, the revolving electrode, every second turn of the engine, wipes over and snaps off the insulated electrode. Thus the points of contact are always kept clean and bright. The igniter is placed directly over the inlet valve to insure a good combustible mixture.

The Olds Gasoline Engine. — Fig. 12-99 shows the carbureter side and Fig. 12-100 the valve construction of the Olds Type *A* gasoline engine. This is an engine of very simple design. Both

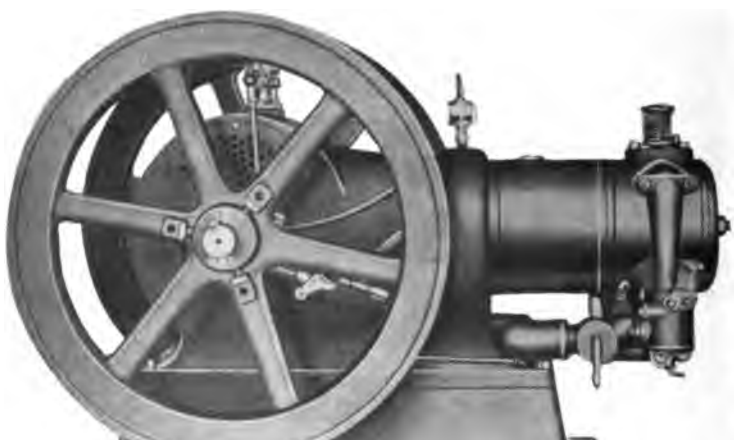


Fig. 12-99. — Olds Gasoline Engine.

valves are of the poppet type, the upper, the inlet valve, being automatic while the exhaust valve is operated by bell crank and push rod. The latter is actuated by a cam. Fig. 12-99 shows the carbureter and its adjusting valve. Until recently the gasoline was fed to the carbureter from a small reservoir kept filled to a certain level by a small plunger pump driven from the shaft. In some of the recent designs this pump has been eliminated, the carbureter getting fuel by simple suction feed.

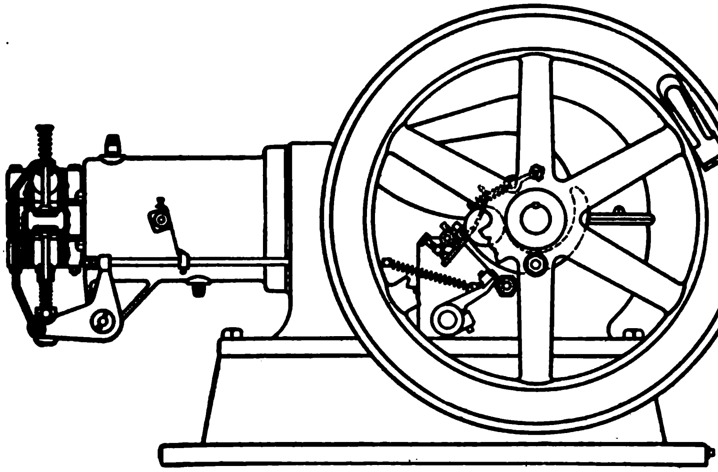


FIG. 12-100. — Olds Gasoline Engine.

The governor, which is of the fly-wheel type, is shown in Fig. 12-100. It operates to keep the exhaust valve open when the speed exceeds the normal, at the same time locking the inlet valve, so that no charge can enter the cylinder. The governor may be adjusted to give speeds varying from 200 to 600, and a change of about 50 turns per minute may be made while the engine is in operation.

The jump spark ignition system is used, which is unusual in small stationary engines. The governor serves also to throw the commutator of the system out of action should the engine fail to take a charge on a miss-stroke.

Type *A* Olds gasoline engines are built in six sizes from 3 to 18 horse-power.

Marine Gasoline Engines. — These engines are of either the

two- or four-cycle type, but in nearly all cases vertical. The maximum power developed in one cylinder, in the ordinary launch motor, is usually about 5–8 horse-power, higher power than this being obtained by the multiplication of cylinders.

Perhaps the majority of engines of this type, especially those of fairly high speed, 600 to 800, are equipped with the jump spark system; many builders, however, favor the hammer break ignition.

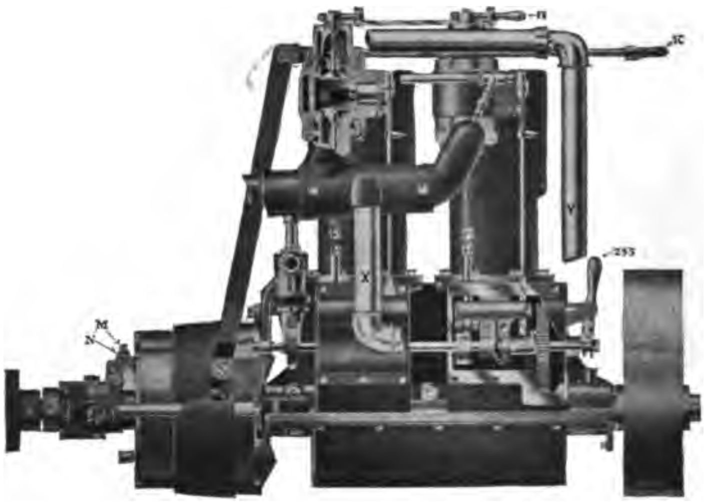


FIG. 12–101. — Strelinger Four-cycle Marine Engine.

Small gasoline boat engines are very rarely fitted with governor control. The speed is changed in most cases with the throttle or the spark, or both, as in automobile engines. A favorite way of checking the speed of small boats is by the use of a reversible propeller, in which case the speed of the engine need not be changed in any way.

The following two types of engines illustrate the general features of the four- and two-cycle marine engines.

The Strelinger Four-cycle Marine Gasoline Engine. — A 10 horse-power engine made by the C. A. Strelinger Company is illustrated in Fig. 12–101 and Fig. 12–102. In this design the heads are cast in one part with the cylinders. The latter are rigidly bolted to the crank case casting. The cam shaft and its drive are enclosed in the crank case. The cams on this shaft

operate the exhaust valve and the trip gear of the make-and-break igniter. By pushing in the lever, marked 255 in Fig. 12–101, at starting, the compression is partly relieved and the spark retarded on all of the cylinders, making starting easier. After starting the lever is returned to its former position. There seems to be no other spark control.

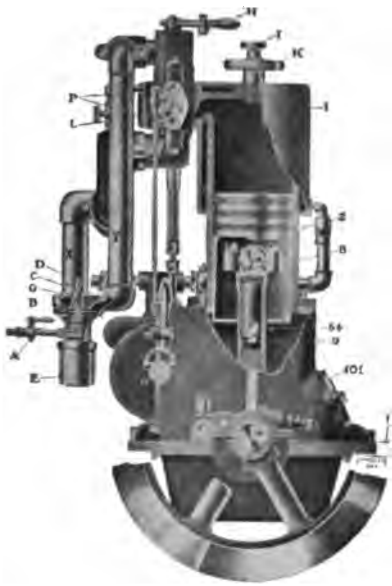


Fig. 12–102.—Strelinger Engine.

The inlet valve is automatic. Above each inlet valve there is a throttle valve whose position is controlled by the lever *H*. The combustible mixture is formed by drawing the air, for the purpose of warming it, from around the exhaust pipe, and through the pipe *X*, Fig. 12–101, through the mixing valve *B*, Fig. 12–102. Here it picks up a certain quantity of gasoline, which is furnished through pipe *A* and the needle valve *C*, and then passes through pipe *Y* to the cylinders. Mixing valve *B* is automatic, adjusting the amount of gasoline to the amount of air passing. The latter of course depends upon the position of the throttle valve and the load on the engine. The mixture is claimed to be uniform through the range of speeds.

The Lozier Two-cycle Marine Engine. — Like nearly all small two-cycle engines, this machine precompresses the charge in the crank case. In Fig. 12–103 * an explosion has just taken place above the piston. The new charge of gasoline vapor and air has just been drawn into the crank case *B* through the opening *A* from the carbureter. The explosion forces the piston down and compresses the charge in *B* to a pressure in the neighborhood of 5 pounds. Near the end of the down stroke the upper edge of the piston uncovers the exhaust port *F*, Fig. 12–104, and the larger part of the burned gases escape. A moment later the inlet port *C*, Fig. 12–105, is also opened, and the new charge enters, being deflected upward by the baffle *G* and driving out the rest of the burned gases. On the next succeeding up stroke the piston closes first the inlet port, next the exhaust port, and compression commences. Ignition takes place when the igniter gear moves one of the electrodes and breaks the contact at *E*, Fig. 12–103. The igniter gear is shown in detail in Fig. 12–106. Normally the outside blade *E* which pivots about *F* is held down by the plunger *D*, so that the lower electrode inside of the cylinder is not in contact with the upper. The rod *B* is moved up and down by an eccentric on the engine shaft. On the upward motion the plunger *D* is forced upward, allowing plunger *M* to raise *E* so that just before the spark is desired the electrodes inside are in contact and the circuit is made. In the meantime the point of the adjustable screw *J* commences to force the trigger *C* to one side until, at the moment the spark is desired, the plunger *D* snaps off suddenly, breaking contact inside by forcing down the blade *E*.

The carbureting device, Fig. 12–107, is simple and effective. The air is pre-heated by passing it around the exhaust pipe. Gasoline enters at *F*, and mixes with the air when the inrush due to the suction in the crank case lifts the automatic valve *B*, flowing out from opening *A* in the seat of the valve. The dial *D* indicates the position of the gasoline valve.

The speed of the engine is controlled by the position of a butterfly throttle valve in the transfer passage, as indicated in Fig. 12–103.

The particular construction of the two-cycle machine above described is known as the two-port two-cycle. In this type the

* Sibley College Thesis of Bayne and Speiden.

Fig. 12-105.

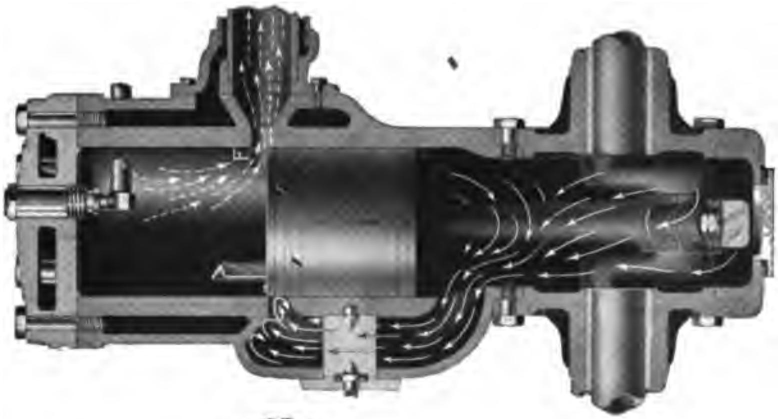


Fig. 12-104.
Operation of Lozier Two-cycle Marine Engine.

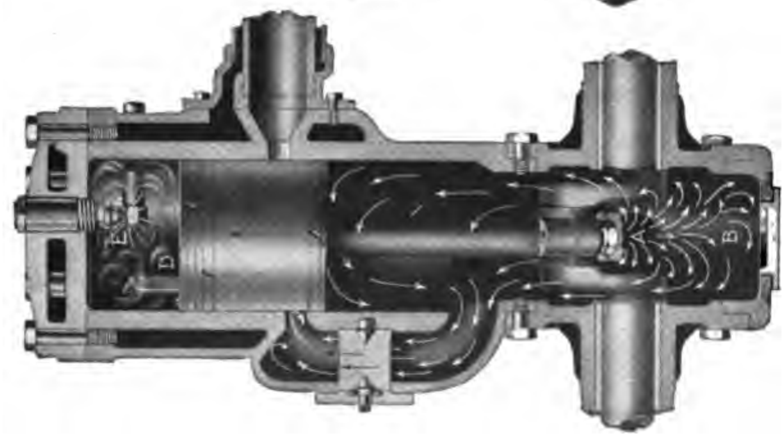


Fig. 12-103.

port admitting the charge to the crank case is at no time covered by the piston. Hence the carbureting device is subject to the pressures generated by possible explosions in the crank case, and if this is of such type as to be hurt by such explosion, it must be suitably protected by check valve. This difficulty is overcome by the use of the three-port two-cycle engine. This is the type of engine in which the piston also controls the inlet port into the crank case, as is clearly shown in Fig. 12-108, which illustrates

Fig. 12-106. — Igniter Gear Lozier Two-cycle Marine Engine.

the construction and operation of the Fairbanks vertical Marine Engine.

The application of the internal combustion engine to the propulsion of vessels other than pleasure launches is becoming of more and more importance. At the present day several rather small sized cargo boats and barges are already fitted with suction gas apparatus and engines. The question has perhaps received the greatest amount of attention in England, where

Crossley Brothers, Thornycroft, and Vickers Sons & Maxim have built suction gas apparatus for marine propulsion.* In this country the Standard Motor Construction Company of Jersey City have lately turned out a marine gas engine which deserves special attention.

FIG. 12-107. — Carbureter Lozier Two-cycle Marine Engine.

The Standard Marine Gasoline Engine. — The following description of this engine appeared in *International Marine Engineering*, September, 1907. The data given refers to the 300 horse-power engine of the motor yacht Standard, but a 500 horse-power engine of the same type for the schooner Northland has already been built.

Figure 12-109 shows a cross-section of the Standard's engine, and Fig. 12-110 a view of the inlet side. The following is the description given:

"These engines have in reality twelve working cylinders,

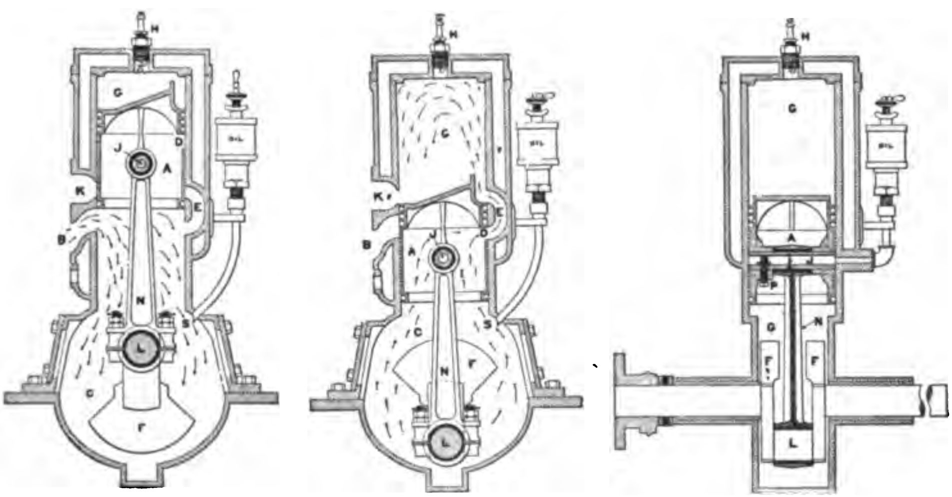


FIG. 12-108. — Fairbank Vertical Three Port Two-cycle Marine Engine.

figured from the point of view of the usual construction of gas engines. The result is an extremely smooth and quiet running

* See an interesting article by A. V. Coster on "Gas Power on Shipboard," in Cassier's Magazine, November, 1907.

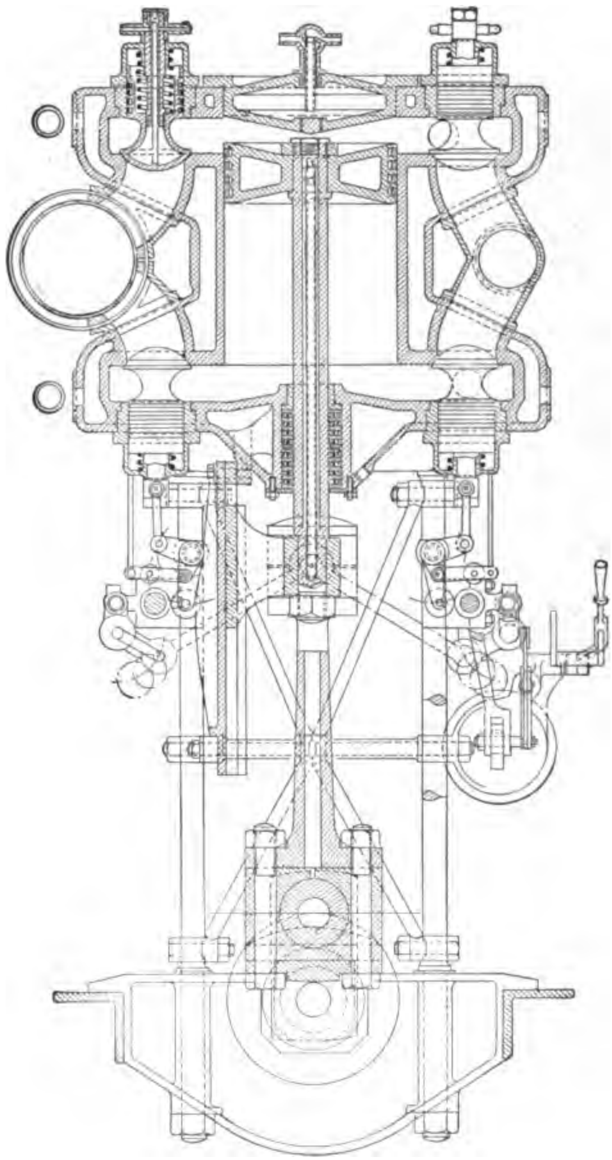


FIG. 12-109. — Cross-section of 300 H.P. Standard Marine Gasoline Engine.

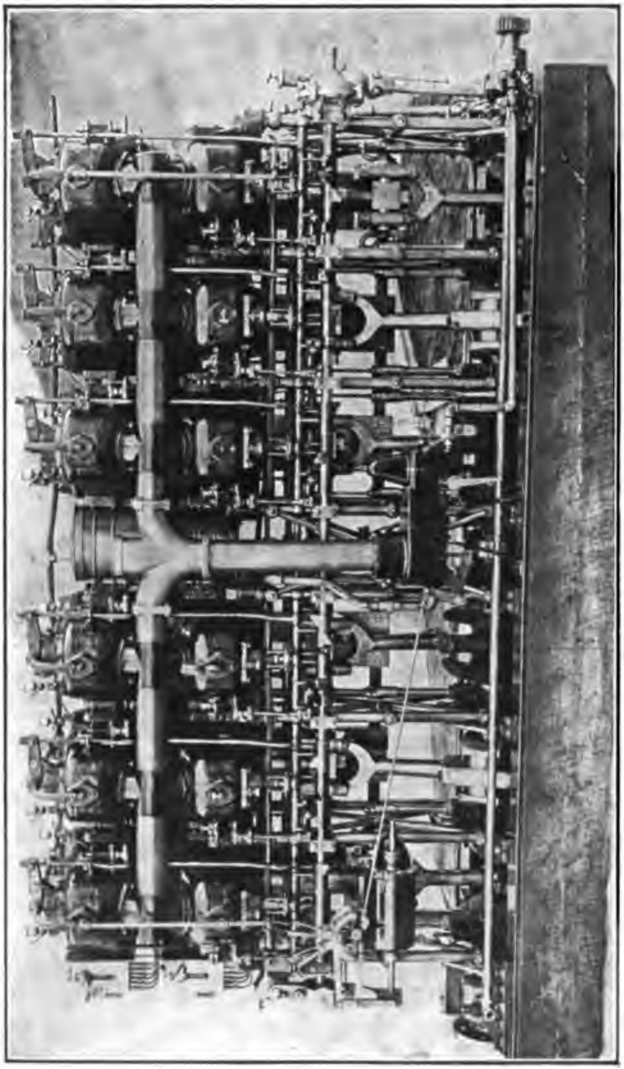


FIG. 12-110. — Inlet Side of 300 H.P. Standard Marine Gasoline Engine.

machine, well balanced, and with the valves so arranged that practically the only vibration noticeable is occasioned by the force feed oil pumps, which give individual feed to every cylinder and bearing. Ignition is of the make-and-break type, while pistons and piston rods are water-cooled, and water flows freely also through the valves. A very noticeable advantage of the double-acting engine is the practical impossiblity of any leakage of gas into the engine room, because the piston, instead of being open to the atmosphere, as in most types of single-acting engine, is operated from a piston rod running through a metallic stuffing-box. The operation of the valve shaft is obtained, as in a large steam engine, by using a small air cylinder; this is also used for turning the engine over.

"The bore and stroke of the Standard's engine are each 10 inches; the weight is 7500 pounds, or 25 pounds per horse-power; the length over cylinders is 8 feet 3 inches; length over all, 10 feet 4 inches; the total height is 5 feet 6½ inches, of which 4 feet 9½ inches come above the center of the shaft; the width of the base is 30 inches; while the greatest width over cylinders is 34 inches.

"The illustrations of the new motor, as shown, serve to confirm the impression of a torpedo boat engine, as the same type of supports, connecting rods, cross-heads, and guides are here used that one is accustomed to in the other kind of engine. The double-acting principle necessitates some radical changes from the usual marine gasoline motor design, such as water-cooling the pistons, connecting rods, valves and other parts exposed to the intense heat produced in cylinders of such large volume. Inlet and exhaust valves are located on the opposite sides of the head, as is customary with many automobile and marine engine builders. The upper and lower ends of the cylinders are alike, with a regular marine engine piston and connecting rod.

"The method of getting the gas into the cylinders is very neatly and simply worked out, as is shown in the inlet side. One branch pipe leads up to the cylinders from the center, and then runs to each end of three cylinders. The spark and throttle control levers are mounted on top of the air cylinder, which controls the travel of the cam shaft. One essential point of difference between the ordinary motor and this motor is that the valves are pulled up instead of being pushed up, that they are

balanced and that they are water-cooled by a stream of water which flows through them constantly. This type of valve has been adopted by the Standard Company for all their large type of motors, and has been found to work exceedingly well. On this side, also, are located the trips which operate the sparkers for all cylinders.

"The exhaust valves are operated from a cam shaft which is a duplicate of that upon the inlet side. These valves are also water-cooled and discharge directly into a large water-cooled exhaust box, from which the gases are led to the muffler. The oil feeds are directly under the exhaust box, with individual leads to distribute the oil to the different bearings. Directly below and behind this are shown the air inlets to the cylinders for starting. In a large motor of this kind the problem of properly cooling it and keeping it well oiled is a difficult one, and the working out of the systems by which these results are accomplished is more complicated than it would be for a small motor, but the owner of a motor of this size could hardly expect to operate his own engine, and in reality the successful working of the motor does not demand a higher order of skill than is to be expected from the engineer of an ordinary triple-expansion steam engine. While the appearance of the motor is complicated, this complication is more apparent than real. Every function of the motor has been carefully thought out and provision made to assure its proper working under all conditions. All Standard motors of large power have been made self-starting and reversing, the same principles which have been found so successful being employed in this latest type: but in the double-acting type both the inlet and exhaust valves are mechanically operated, as against the exhaust valves only in the other self-starting and reversing motors. Another important difference is that in this type all the cylinders are cast separately, and the motor is really two three-cylinder motors coupled together for purposes of economy in construction and replacement, should such be necessary.

"The self-starting is accomplished by compressed air being admitted on one end of three cylinders by a special set of cams. The cylinders turn the motor until it takes up its cycle upon gasoline. The cam shafts which actuate all valves are so made as to move longitudinally on their axes, and to bring appropriate cams into action for either direction. As the physical labor required to

perform this operation would be considerable in a motor of this size, an air cylinder is added to perform this work, so that the only manual labor in connection with running the motor is that in operating the two handles shown on the left side of Fig. 12–110. One lever controls the position of the cams, whether for ahead or astern, while the other controls the spark and throttle. In this motor the necessity of a fly-wheel is entirely done away with, and operating at any speed, from maximum to minimum, little or no vibration is felt. The motor is mounted upon a base of angle iron with castings for each individual bearing, and has the usual marine collar thrust bearing."

Fig. 12–111. — Engine of 1907 Franklin Car.

The Automobile Gasoline Engine. — It would be beyond the scope of this book to enter into any extensive notice of automobile engines. For that reason only a few illustrations are given to show the various methods of placing the valves and operating them, the methods of cooling the cylinder, cylinder and frame construction, etc. In general the four-cycle engine monopolizes the automobile field, there being but one or two makers who use two-cycle engines. For light cars the horizontal opposed two-cylinder type of engine is often employed, but the heavier cars universally use four- or six-cylinder vertical engines. Regarding the position of the valves, some makers place both inlet and exhaust valves in the head. This type is best exemplified by the Franklin Engine of 1907, Fig. 12–111. This construction is of advantage because it does away with all pockets in the combus-

tion chamber. In this engine both valves are mechanically operated. Some makers use the automatic type of inlet valve,

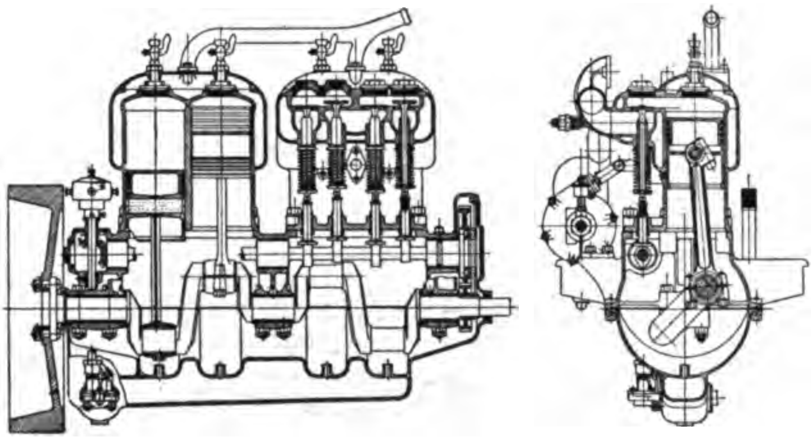


FIG. 12-112. — Continental Automobile Engine.

which in such cases nearly always opens downward. This valve, while it permits of simpler engine construction than the mechanically operated valve, does not give as good service, and for that reason is not as frequently used as formerly. In the Continental Engine,* Fig. 12-112, both valves are placed at one side of the cylinder and both are mechanically operated, opening upward. In the 18 horse-power Horch engine, Fig. 12-113 † the inlet valve is placed in the head, the exhaust valve at the side. The 35 horse-power engine of the same maker, Fig. 12-114 ‡ has

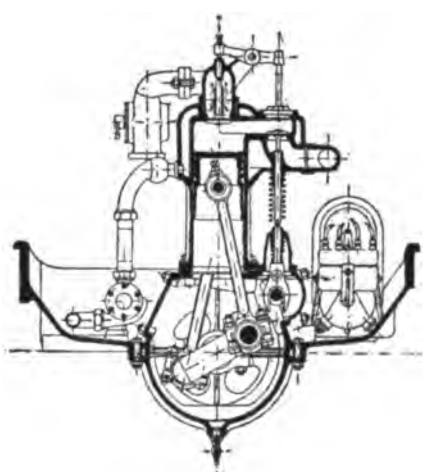


FIG. 12-113. — 18 H.P. Horch Automobile Engine.

* The Cycle and Automobile Trade Journal, Oct. 1, 1907.

† Gasmotorentechnik, April, 1906. ‡ Gasmotorentechnik, April, 1906.

both valves placed at one side, but the inlet valve over the exhaust valve. Both are mechanically operated. The Moore engine, Fig. 12–115,* shows the valves upon opposite sides, both mechanically operated. The 1908 model of the Franklin engine finally shows a type of valve gear different from any of those so far described in that the main exhaust and the inlet valves are combined in one concentric valve placed in the head. This, as may be seen from Fig. 12–116, results in an ideal form of combustion chamber. The outside, larger, valve is the inlet valve. The large inlet area improves the volumetric efficiency. This, however, is in large part also made possible by the fact that the greater part of the hot burned gases escapes through the auxiliary exhaust valve which opens when the piston is near the lower dead center. The remainder of the gases passing out through the inner valve at the top during the up stroke of the piston does not tend to heat the valve very much.

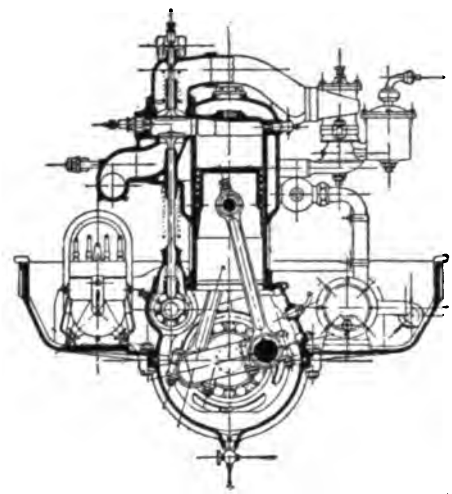


FIG. 12–114. — 35 H.P. Horch Automobile Engine.

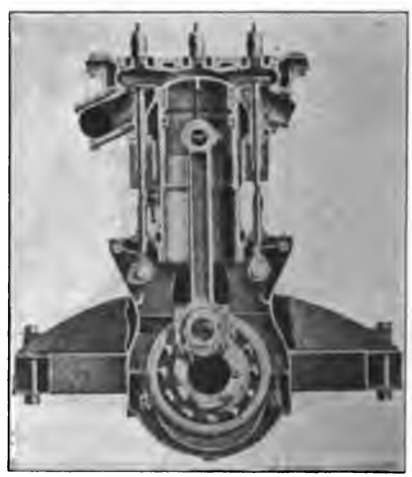


FIG. 12–115. — Moore Automobile Engine.

Of the automobile engines

* Cycle and Automobile Trade Journal, April, 1906.

illustrated, the Franklin engine shows the construction employed for air-cooled cylinders; all of the other engines are water-cooled.

4. Oil Engines. — The term "oil engines" usually refers to engines using kerosene, crude oil, or any one of the so-called distillates. All of these fuels are more difficult to vaporize than gasoline and the formation of the proper fuel mixture is therefore generally a less simple process. Some of the types use a special vaporizer, in others the fuel is sprayed directly into the combustion chamber or into an extension of it.

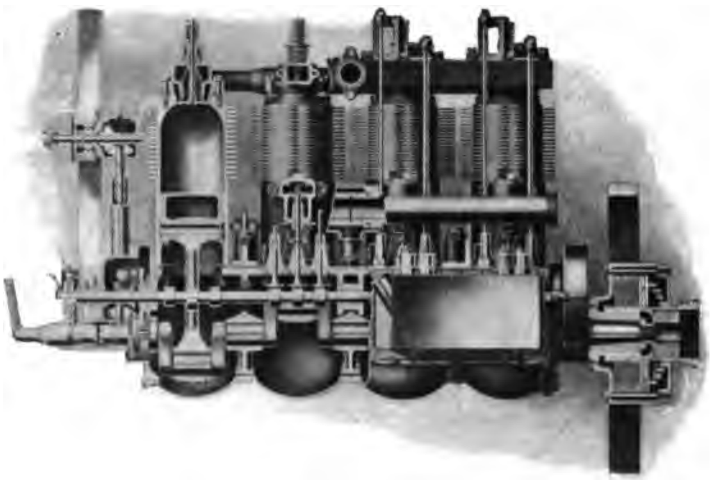


Fig. 12-116. — 1908 Model, Franklin Engine.

Liquid fuel engines in general labor under the disadvantage as compared with gas engines in that the limit of power in a single cylinder is reached much earlier. The reason for this is that as the cylinder volume grows, the difficulties of forming and maintaining a uniform fuel mixture increase very rapidly, and the difficulties of proper ignition increase correspondingly. The result is, at least as far as present practice is concerned, that while the economic limit in one single-acting cylinder using gas is in the neighborhood of perhaps 400 horse-power, the limit in single-cylinder single acting oil engines is not over 200 horse-power. Hence the subdivision of the power required among several cylinders commences much sooner in oil than in gas engines.

The most important point of distinction among the various

oil engines is probably the way in which the combustible mixture is formed. We distinguish the following methods:

(*a*) The fuel is forced directly into the combustion chamber at the end of or during the compression stroke. Usually a spraying nozzle is employed.

(*b*) The fuel is mechanically atomized and sprayed into the current of air on the suction stroke.

(*c*) The fuel is sprayed into a vaporizing chamber connected to the cylinder without interposition of a valve.

(*d*) The fuel is vaporized by the agency of heat in a separate vaporizer. The air passing through the vaporizer forms the fuel mixture and this enters the engine already prepared.

The following descriptions of various oil engines will illustrate the various methods outlined:

Among the most important oil engines found on the American market are the following: The Hornsby-Akroyd kerosene engine, a machine of English origin made by the De La Vergne Machine Company, of New York; the De La Vergne two-cycle oil engine, brought out by the same company within the last month or two; the Mietz & Weiss oil engine made by the A. Mietz Iron Foundry & Machine Works of New York; and the American Diesel engine manufactured by the American Diesel Engine Company. It should be stated that many of the engines described under the head of small and medium sized gas engines can be run on oil by using suitable vaporizers. Thus the Fairbanks-Morse Company and several other makers furnish vaporizing attachments by means of which their engines may be successfully run on kerosene, crude oil, or distillate.

The Hornsby-Akroyd Oil Engine. — Figs. 12-117 and 12-118 show a longitudinal and a transverse section respectively of this important oil engine, while Fig. 12-119 shows the general appearance of a single-cylinder engine.* The Hornsby engines built in this country are all horizontal machines of the four-cycle type. The exhaust and inlet valves are of the poppet type and located in a valve box at the side of the cylinder, Fig. 12-118. They open upward and are operated by levers passing under the cylinder by means of cams on the half-time shaft. The exhaust cam is so designed that on starting the compression can be relieved

* Catalogue of the De La Vergne Machine Co.

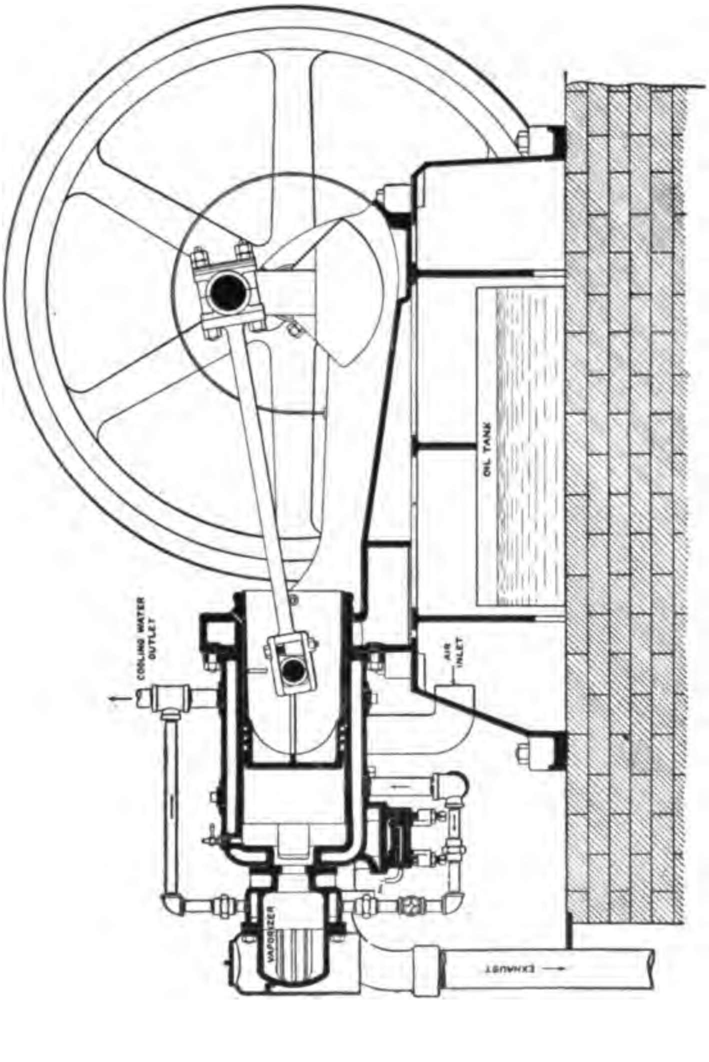


FIG. 12-117. — Cross-section Hornsby-Akroyd Oil Engine.

by shifting the cam on the shaft. The supply of oil is taken from a tank in the base of the machine by a pump operated from the lay shaft, and forced into the vaporizer chamber A, through a spray nozzle. The governor is of the fly-ball type and controls the speed by dividing the constant quantity of oil furnished by the pump into two parts, one of which, in proportion to the load, enters the nozzle, the other part flows back to the tank. The nozzle and overflow valve are shown in greater detail in Chapter VIII.

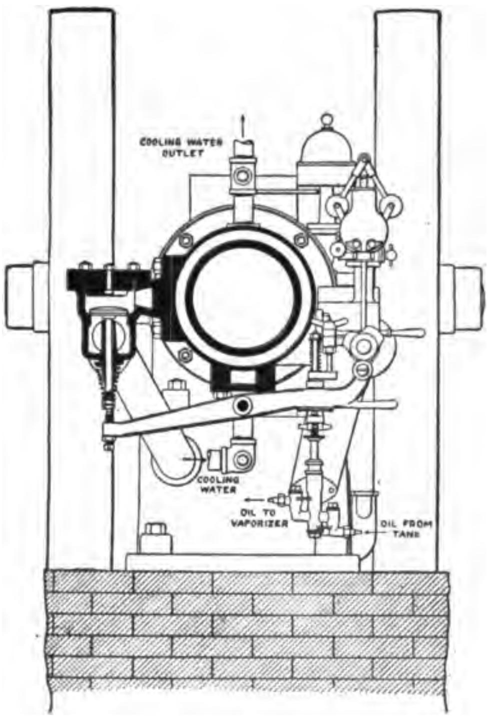


Fig. 12-118. — Transverse Section Hornsby-Akroyd Oil Engine.

The vaporizer chamber is furnished with internal webs to increase the vaporizing surface, and is protected against radiation on the outside by a hood.

To start the engine the vaporizer is first heated by a lamp which constitutes a part of the equipment. After the chamber

is hot enough, a few quick strokes of the pump by means of the hand lever, Fig. 12-118, while the engine is being turned over in the normal direction, usually suffices to start it. After starting, the heat of combustion is enough to keep the vaporizer at a dull red heat and to explode the charges regularly, so that no special igniter is required.

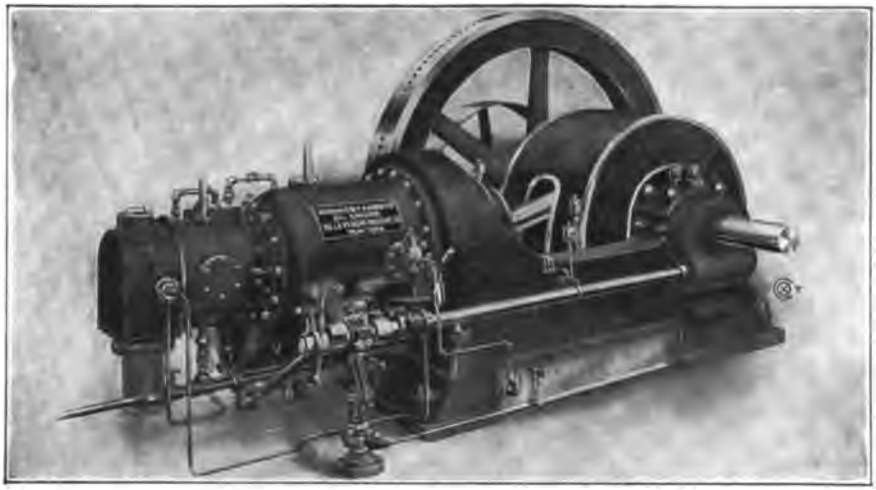


FIG. 12-119. — Single Cylinder Hornsby-Akroyd Oil Engine.

The method of operation may be explained as follows: On the suction stroke of the piston the pump injects oil into the vaporizer. This is almost instantly vaporized, but as yet the mixture is not explosive because the vapor is mixed mainly with burned gases which remain from the previous explosion. On the return stroke, the piston compresses the air and forces a part of it into the vaporizer. It is possible that some time during the compression stroke the vapor may commence to burn in the vaporizer, but the flame does not strike out because the velocity of air flowing in through the narrow neck of the vaporizer is greater than the velocity of flame propagation. Near the end of the compression stroke the reverse takes place, the flame strikes out with explosive force and drives the piston forward on the expansion stroke. This is followed by the exhaust stroke, after which the operation is repeated.

These engines have given very satisfactory service and many of them are in use for a variety of purposes. They are made in sizes from 2½ to 125 horse-power, those above 32 horse-power being of the two-cylinder type illustrated in Fig. 12–120. This design has lately been modified in that the two overhanging fly-wheels have been replaced by a single wheel placed next to the belt pulley between the two cylinders.

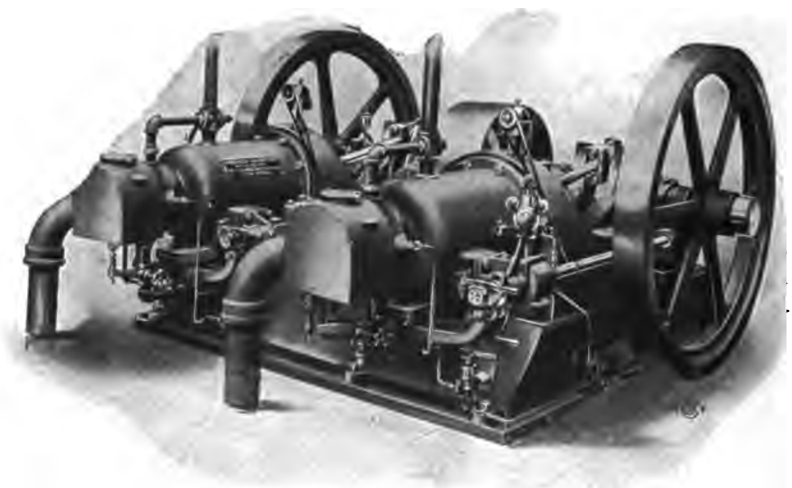


FIG. 12–120. — Twin Cylinder Hornsby-Akroyd Oil Engine.

The De La Vergne Two-cycle Oil Engine. — This is a vertical machine recently brought out by the De La Vergne Machine Company and described in Power, November, 1907, to which the following illustrations are due:

Figure 12–121 shows that the cylinder is of the ordinary two-cycle construction, the piston controlling the inlet and exhaust ports. The vaporizer chamber V is formed by an extension of the cylinder, but, in contradistinction to the Hornsby engine, no contracted neck is used opening into the vaporizer. On the down stroke of the piston the air compressed in the crank case rushes into the cylinder as soon as the inlet port is uncovered. On the up stroke only air is compressed, the oil not being injected by the

pump through the nozzle N until near the upper dead center. The oil instantly vaporizes and burns, ignition being produced by the hot vaporizer walls. Of course the vaporizer must be

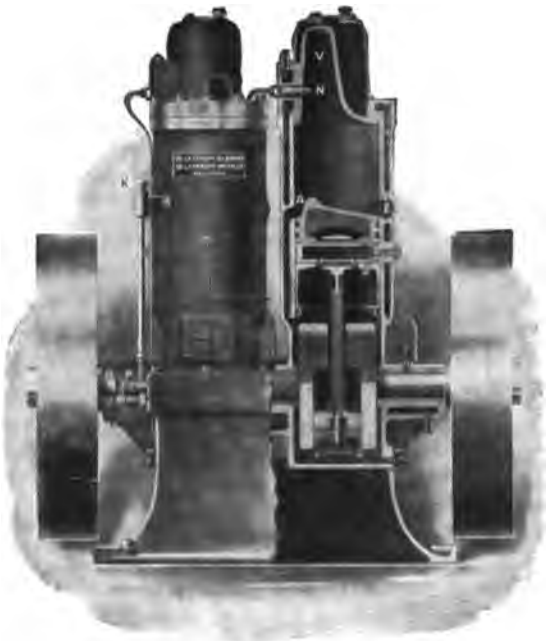


FIG. 12–121. — De La Vergne Two-cycle Oil Engine.

externally heated to start the engine. The method of operation outlined has the advantage that since only air is compressed no pre-ignition can take place.

The details of the spray nozzle N are shown in Fig. 12–122.

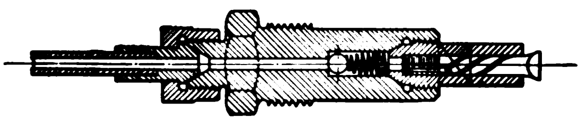


FIG. 12–122. — Spray Nozzle De La Vergne Two-cycle Oil Engine.

They are so simple as to hardly require explanation. The ball check shown serves to protect the oil pipe against the force of the explosions.

The engine is governed by suiting the quantity of oil to the load. This is done by putting the pump plunger under control of the governor in the fly-wheel. Power gives the following description of this device (see Fig. 12–123):

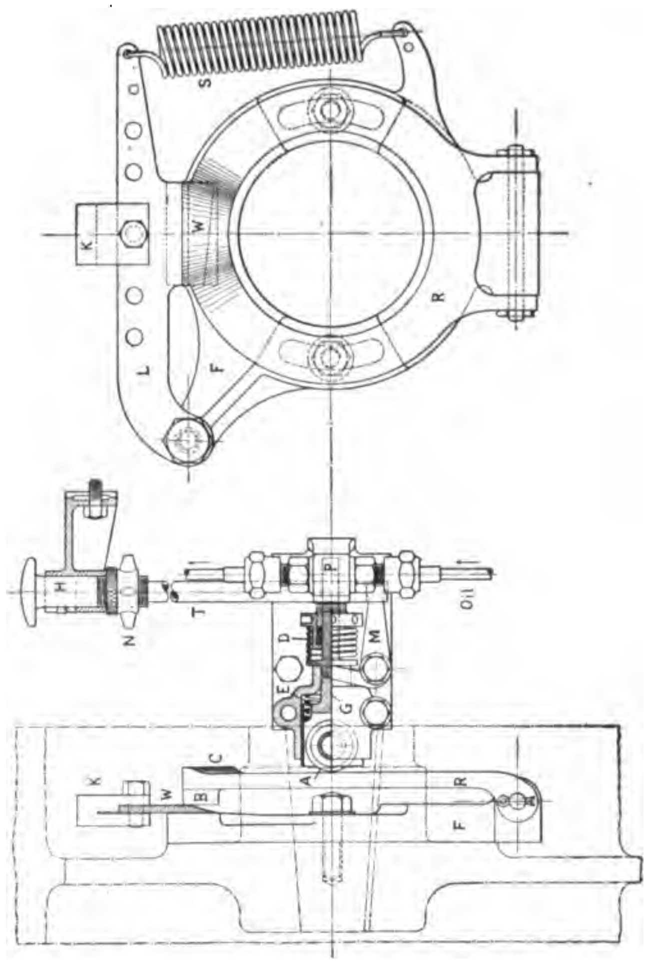


Fig. 12–123. — Governor of De La Vergne Two-cycle Oil Engine.

"The frame *F* is fastened with two studs concentrically to the inside of the fly-wheel, and to the frame is pivoted a cam ring *R* which has on the fly-wheel side a projection *B*, and back of that

the cam projection C, which lifts the roller A once each revolution of the fly-wheel, thereby actuating the oil-pump plunger P. The length of stroke imparted to the plunger is determined by the lever L, pivoted to the frame. A wedge W on the lever L separates the cam ring from the frame F. When the engine is started the governor does not come into action until normal speed is approached. A decrease in load will cause the speed to increase, when the centrifugal force of the counterweight K will overcome the tension of the spring S, moving the weight outward and thereby withdrawing the wedge W. The spring E keeps the roller A in contact with the cam ring R, and when the wedge is withdrawn the buffer G cannot actuate the pump plunger. The knob H is used for injecting oil at the start, and the lock nuts N serve for limiting the stroke of the pump. The two concentric slots in the frame F allow for adjustment when it is desired to greatly change the normal speed of the engine. For smaller speed variations the spring S and the weight K can be shifted in the holes of the lever L."

At present two sizes of this engine are made. That illustrated is of 15 horse-power, the single cylinder type gives 7½ horse-power. In each case the cylinders are 7 inches by 7.5 inches, the normal speed being 450 r.p.m.

The Mietz & Weiss Oil Engine. — This is a very successful engine which has been on the market for some years. While it operates on the ordinary three-port two-cycle principle, it embodies some unusual features. Fig. 12–124 shows the horizontal type in cross-section, and Fig. 12–125 its general appearance in elevation. From Fig. 12–124 the operation of the engine must be clear without much further explanation.

The fuel pump P takes the oil from a tank mounted on the crank case and injects it into the cylinder just after the piston has covered the exhaust port E. The oil falls on the projection V and is instantly vaporized by heat from the hot bulb I, and from the cylinder head, which, as will be noted, is not water-jacketed. Ignition is produced by means of the bulb I, which acts very much like the ordinary hot tube. A kerosene or oil lamp, shown in position, serves to heat the bulb before starting. Five minutes is usually sufficient for this operation.

The unusual feature of this engine consists in the cooling

water arrangements. Instead of circulating the water, as is ordinarily done, it is allowed to evaporate in the jacket, being kept at a constant level by a float valve. The vapor formed collects in the dome *S*, Fig. 12–124, and is by means of the bent pipe shown led to the intake port of the cylinder where it mixes with the air. There are several excellent reasons for this arrangement in oil engines. The water vapor helps to form a mixture of high specific heat, thus reducing the danger of pre-ignition and allow-

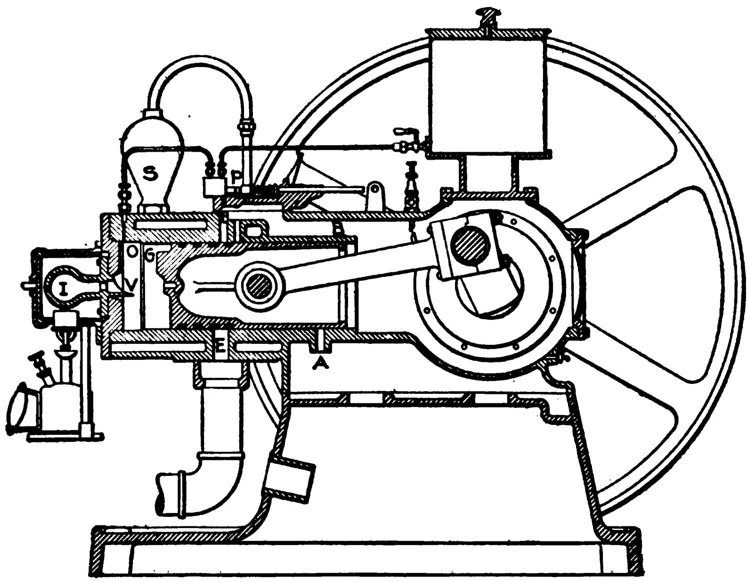


FIG. 12–124. — Details of Mietz & Weiss Two-cycle Oil Engine.

ing of higher compression pressures. The principle, however, is not new, since Bánki and perhaps several others before him have used it. It is also claimed that the vapor assists in the prevention of carbon deposits. The method further has the general advantage that but little jacket water is used compared with the ordinary method of operation.

The method of governing this engine is not clearly shown, but it consists in adjusting the stroke of the fuel pump to suit the load by means of a shifting eccentric on the main shaft.

The stationary Mietz & Weiss engine is built in single-cylinder

units from 1 to 30 horse-power, while the 40 and 60 horse-power sizes are twin engines.

Fig. 12-125. — Mietz & Weiss, Two-cycle Oil Engine.

This firm also builds vertical marine engines, the operation of which is the same as that of the horizontal machine except that the water injection scheme is not used.

The Diesel Engine. — The Diesel engine is to-day built by a number of firms in Europe and by the American Diesel Engine

Company in this country. In all cases the various constructions are of the vertical four-cycle type, but European builders favor the open *A*-frame, while the American makers have adopted the enclosed box frame.

Figure 12–126 gives a general idea of the appearance of the American Diesel engine, while Fig. 12–127 shows the construction.

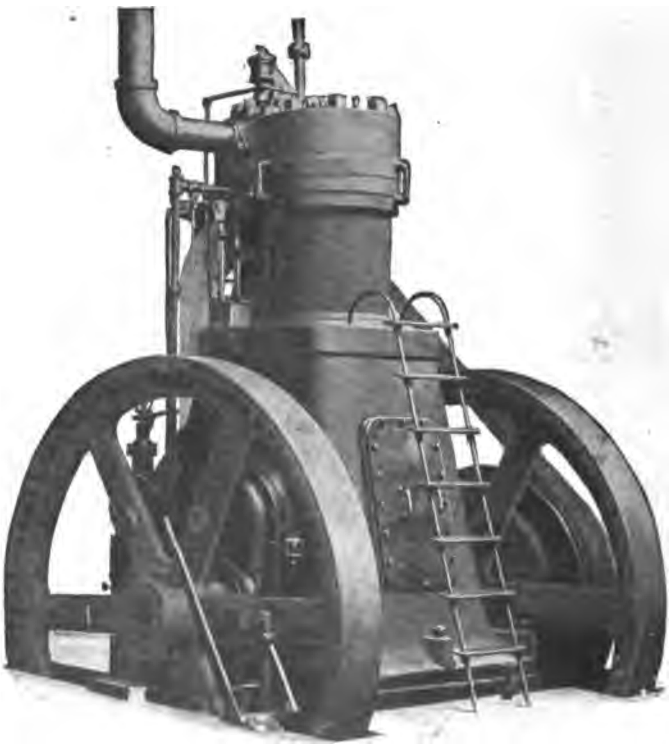


Fig. 12-126. — American Diesel Engine.

The massive self-contained build of this machine, necessitated of course by the high pressures occurring in its cycle of operation, is very noticeable. The method of operation of this machine has already been explained in Chapter XI, to which the reader is referred.

The valve construction of this engine is very simple, as shown in Fig. 12–128. The exhaust valve opens upward, the inlet valve

downward; both are located in a small chamber at the side of the cylinder. The fuel injection valve *I* is opened by the bell-crank *B*, which is operated by a cam on the lay shaft, and closed by a helical spring *S*. Surrounding the spindle of the injection valve

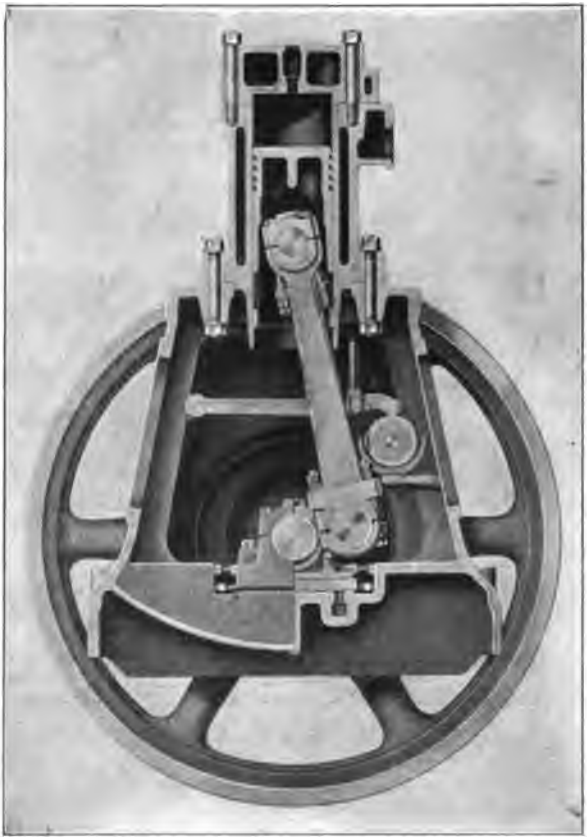


FIG. 12-127. — Cross-section American Diesel Engine.

are placed the atomizing arrangements by which the oil is very finely divided through the agency of highly compressed air, as soon as *B* opens the valve. The stroke of the lever *B* is uniform, hence the injection valve always opens to the same amount and for the same length of time, whatever the load on the engine. To govern the speed, the governor controls the stroke of the

pump which furnishes the oil to the valve. The lower the load on the engine, the later in the stroke of the oil pump does the delivery of oil to the injection valve commence. One method of doing this is explained in Chapter XIV, under Details of Governors.

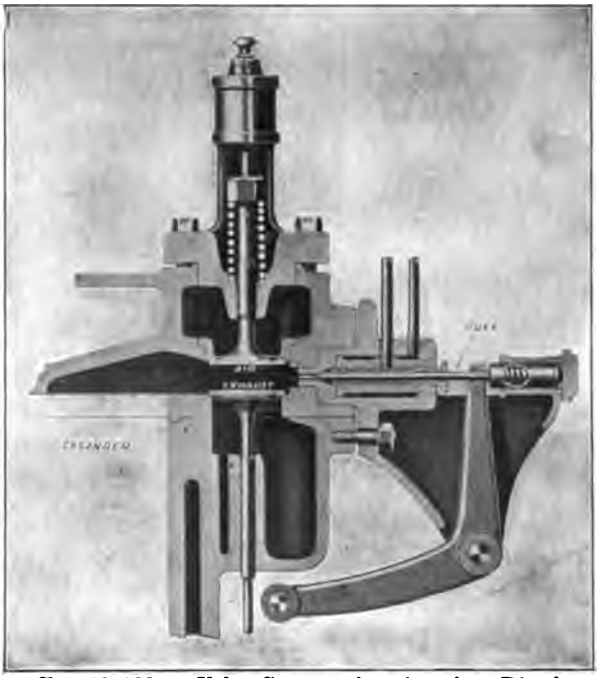


FIG. 12-128. — Valve Construction, American Diesel Engine.

The American Diesel engine is built in sizes from 75 to 450 B. H. P., mostly in three-cylinder units.

The Priestman Oil Engine. — This English machine, Fig. 12-129,* is mentioned here because the means used for forming the combustible mixture are different from those so far described. The engine is of the horizontal single-acting type. The exhaust valve is mechanically operated by an eccentric on the shaft. The same eccentric rod also operates a small air pump *e* which keeps up an air pressure of from 30 to 40 pounds on top of the oil in the

* Güldner, p. 118.

supply tank *b*. The inlet valve *c* is automatic. On the suction stroke the piston draws a charge of finely divided oil, mixed with air furnished by the spray maker, into the vaporizer *a*, together with a large quantity of auxiliary air to form the proper combustible mixture. The vaporizer, heated at the start by external means, is during operation kept by the exhaust gases at sufficient heat to completely vaporize the oil before it passes out on its way to the inlet valve. Thus the mixture reaches the cylinder completely prepared. The spray maker and vaporizer are explained in greater detail in Chapter VIII.

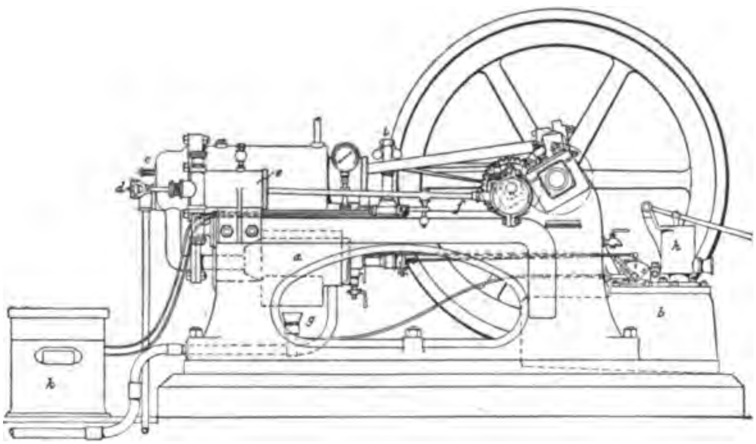


FIG. 12–129. — Priestman Oil Engine.

The cylinder operates on the ordinary four-cycle principle; ignition is by electric spark. In the later Priestman engines a small amount of water from the jacket is admitted to the cylinder each cycle, after the principle of Bánki.

The Fairbanks-Morse Crude Oil Vaporizer. — In the fourth type of vaporizer the oil is not preliminarily sprayed or atomized, as is done in the Priestman vaporizer, but simply vaporized by the heat of the exhaust gases. The piston then generally draws a part of the necessary air through the vaporizer and saturates this with the oil vapor. The rest is added to form the proper mixture just before the inlet valve. Of this type is the "Economist" retort, already described in Chapter VIII. Another type is that made by the Fairbanks-Morse Company, and illustrated

in Fig. 12–130. G is the main vaporizer chamber. The oil pump furnishes oil through the pipe F to the reservoir R on top of the chamber. The regulating valve T returns the excess pumped to the main supply through the pipe O. From R the oil is allowed to trickle slowly downward over surfaces heated by the exhaust gases. These come in through the pipe N, but the volume entering G is controlled by the position of the valve E, which sends a part into G, the rest directly out through X, depending upon the demands of the vaporizer. Air enters the chamber through C.

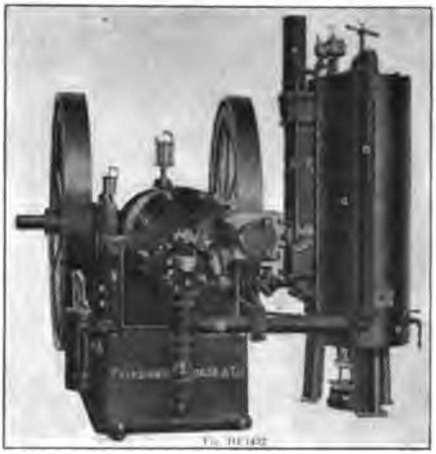


FIG. 12–130. — Fairbanks-Morse Engine with Crude Oil Vaporizer.

On its way upward it saturates itself with oil vapor and finally flows out through B to the engine. The auxiliary air supply is furnished through A.

Any vaporizer of this type labors under the disadvantage, already mentioned in the case of the "Economist" retort, that not all of the oil can be utilized. As the vaporization proceeds, the oil gets heavier, the amount of vapor evolved grows less at the temperature maintained, and the useless residue must finally be drawn off. In the apparatus above described provision for this has been made through the drain cock D. W is a heating lamp to heat the vaporizer on starting.

5. THE ALCOHOL ENGINE. — The points of difference between the alcohol engine and any other gas engine have already been

mentioned in Chapter VIII. In constructive details these engines do not differ from the rest except that the compression is carried higher than in other liquid fuel engines. The main feature distinguishing them is in the arrangements for forming the combustible mixture, and the various expedients adopted have been thoroughly discussed under the head of vaporizers in the chapter mentioned. Recent experiments have disproven the old statement that an alcohol engine cannot be started from the cold. Much depends upon the position of the carbureter with reference to the inlet valve ports, and after paying due attention to this point, a gasoline automobile engine has been successfully operated on alcohol without change. This bears out the experience of the Deutz Company in whose alcohol engines only a spray nozzle placed very close to the inlet valve is used. An ordinary gasoline carbureter can be made to act in somewhat the same way. American practice regarding alcohol engines is for obvious reasons somewhat behind that of Europe, but the indications are at present that this condition will not long exist.

CHAPTER XIII

GAS ENGINE AUXILIARIES: IGNITION, MUFFLERS, AND STARTING APPARATUS

Ignition. — There are four methods of igniting the combustible charge in a gas engine. Some of these are still in use, others belong to the period of gas-engine development. These methods are the following:

1. Ignition by an open flame.
2. Ignition by hot tube.
3. Ignition by heat of compression, and
4. Ignition by electric spark.

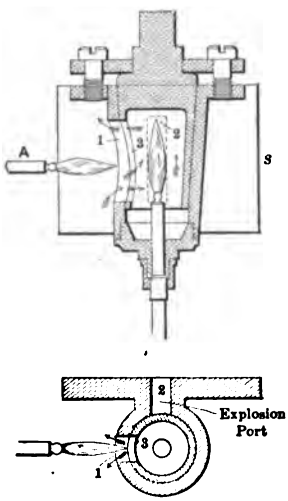


FIG. 13-1. — Barnett's Ignition Cock.

Ignition by open flame is practically obsolete, and ignition by hot tube is also fast falling into disuse. As a matter of fact, except for the Diesel engine, the Hornsby-Akroyd and a few others, which ignite the charge or fuel by heat of compression, the method of igniting the charge by the electric spark is to-day the means most generally employed.

1. **Ignition by Open Flame.** — This method has been superseded probably because of its occasional failure and the obvious danger connected with its use under certain conditions.

The simplest arrangement of this type is Barnett's ignition cock, Fig. 13-1.* This consists of a hollow plug which works in a shell having two ports, 1 and 2. The former opens to the atmosphere and communicates with an outside flame, A, the latter opens into the cylinder. The port 3 in the plug is of such a size that it may communicate either with port 1 or 2. but never

* Clerk, The Gas and Oil Engine, p. 207.

with both at the same time. Inside the plug is placed a gas jet as shown. Gas should be admitted to this at such a rate that the flame burns inside of the plug and not out through the ports 3 and 1. The proper size of the ports has much to do with the proper admission of air to the inside of the plug to keep the flame alive, as shown by the arrows. If now the plug is quickly turned through 90 degrees, making the port 3 register with port 2, enough air is contained in the plug to keep the flame burning during the interval, and the combustible mixture, entering the plug much as the air enters it in the first position, is instantly ignited. Here again the proper size of the ports is of importance, for if the mixture cannot circulate through the plug, it must reach the flame by diffusion. The chances are that in that time the flame has died out and ignition fails. The flame is blown out by the force of the explosion, or dies out for lack of air, but is immediately relighted by the outside flame, *A*, when the plug returns to its former position.

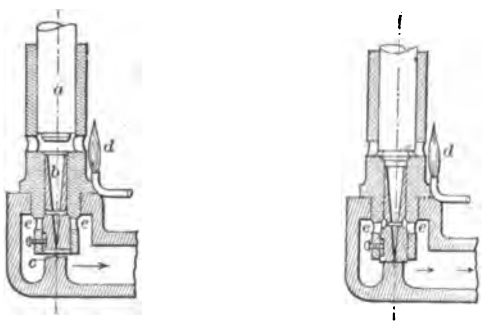


FIG. 13-2.—Koerting Igniter.

Barnett's scheme was open to the objection that since the flame chamber in the plug was always under atmospheric pressure, the combustible mixture, if compressed to any extent, might extinguish the flame by sudden inrush when communication was established. Several methods to obviate this were invented, notably by Otto and by Clerk. The latter differs from the former in that it will operate at higher engine speeds; Otto's scheme failing at comparatively low speeds. A very simple solution of the problem is shown in Koerting's igniter, Fig. 13-2.* The plug, *b*, see left-hand section, is practically a divergent nozzle. The pressure of the combustible mixture during compression raises the plug, and some of the mixture, escaping through the fine opening, *c*, and expanding through *b*, is ignited by the open flame, *d*. At the instant the piston reverses, the plug is forced down by the plunger,

* Schöttler, Die Gasmaschine.

a, closing both the top and the bottom openings. But the mixture contained in the nozzle, *b*, keeps on burning until, at the instant the side openings, *e*, are freed, the flame strikes into the cylinder and ignites the charge. The right-hand section, Fig. 13–2, shows the ignition position.

2. **Ignition by Hot Tube.** — The simplest form of the hot tube ignition apparatus has already been shown in Fig. 11–5 Chapter XI. It consists merely of a small tube 3 or 4 inches long, of steel, porcelain, or platinum. The open end of this tube is in communication with the combustion chamber, the other end is closed. The tube is kept red hot for a certain part of its length, generally by means of a Bunsen burner. A chimney surrounds both burner and tube to prevent loss of heat by radiation as far as possible.

FIG. 13–3.—Adjustable Hot Tube.

The action of the hot tube may be explained as follows: At the end of the exhaust stroke the tube is filled with burned gases, and these are not replaced by the fresh mixture even at the end of the next suction stroke, because the time available is too short for diffusion. During the compression stroke the burned gases are being compressed into the closed end of the tube and are followed up by the fresh mixture. But no explosion follows even if this mixture reaches the red-hot part of the tube as long as the velocity of flame propagation out of the tube is less than the velocity of the fresh charge into the tube. When the former becomes greater than the latter, which happens at or just before the piston reaches the dead center, the flame shoots out and ignites the charge. It is plain, however, that in the device shown in Fig. 11–5 the position of the hot zone along the tube must be about right or pre-ignition may result. Want of adjustment in this arrangement has led to the improved hot tube shown in Fig. 13–3. In this case the position of the hot zone along the tube may be varied as shown, thus giving some control over the time of ignition.

In order to completely control the time of ignition the ordinary hot tube has been perfected by the addition of a so-called timing valve.

In Fig. 13–4, *G* is the tube kept hot in the ordinary way. Timing valve *E* is normally kept closed by the coil spring *C*. At the proper time in the cycle the ignition cam (not shown), through the link, *B*, and the bell-crank, *A–D*, compresses the spring, *C*, and opens the valve. Ignition then ensues. The valve is kept open during the expansion and exhaust strokes. The time of ignition may be changed by changing the position of the cam.

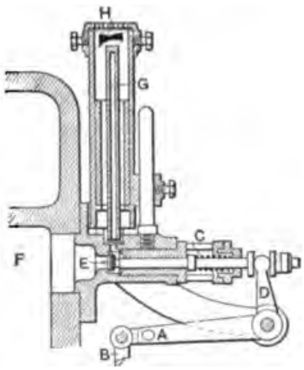


Fig. 13–4. — Hot Tube with Timing Valve.

Timing valves are open to the objection that they are very difficult to keep in shape under the high temperatures occurring. To avoid the use of the small valve in the cylinder, Koerting has used the scheme shown in Fig. 13–5; *a* is an open hot tube made of porcelain. In it there is placed the small platinum tube, *c*. During compression some of the mixture escapes through *c* and the valve *k*. When ignition is desired, *k* shuts the exit and the flame in *a* strikes back into the cylinder.

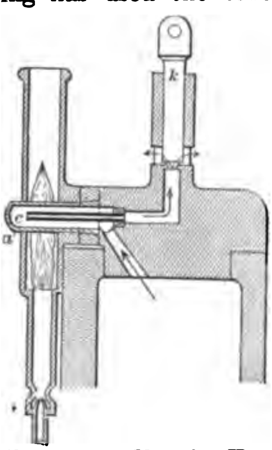


Fig. 13–5. — Koerting Hot Tube Igniter.

Hot tubes may be from two to four inches long, and from one-quarter to one-half inches internal diameter. They may be made of steel, platinum, or Porcelain. Porcelain is best because cheap and nearly indestructible by heat.

The hot tube finds application in small and medium sized stationary machines only. It is fully as cheap to operate as electric ignition and just as certain.

In large machines this method of ignition is not as satisfactory, because the ignition itself is hardly sharp enough for the large volume of gas, and because in many cases the distance

the flame has to strike is too great. Care should be taken to place the tube at the proper point, *i.e.*, where a good mixture at the opening of the tube is insured, and where the opening cannot be clogged by oil or water.

3. **Igniting the Charge by Heat of Compression.** — This method of igniting the charge is practically limited to liquid fuels and is carried out in several ways.

(*a*) Only air is compressed to a very high degree so that its temperature is high enough to ignite the fuel as it is injected at the beginning of the working stroke. This is Diesel's method.

(*b*) The charge may be ignited by means of a hot bulb or chamber connected to the combustion chamber proper by a narrow neck or opening. There are two modifications of this method. In the one the fuel is injected into the combustion chamber on the suction stroke. During the next stroke the mixture is compressed into the hot bulb and ignites. The combustion, however, is confined to the bulb until, near the end of the compression stroke, the velocity of flame propagation exceeds the velocity of gases entering the narrow neck of the bulb, when the flame strikes out and general ignition ensues. The action of the bulb is therefore very similar to that of the open hot tube. The main difference is that after the bulb has been externally heated at the start, the heat of compression soon keeps the walls of the bulb at a sufficiently high temperature so that the external flame can be extinguished.

In the second modification, the hot bulb or chamber is used at the same time as a vaporizer. Thus in the Hornsby-Akroyd engine, the fuel is injected into the chamber by a pump at the beginning of the suction stroke. The piston draws nothing but air on this stroke, which air is partly forced into the bulb on the return stroke. Here it mixes with the oil vapor, which formed, due to contact with the hot walls, and while combustion may ensue it cannot be general because hardly enough oxygen is present in the bulb or vaporizer. Near the end of the compression stroke, however, the flame strikes out, and the combustion becomes explosive. As in the former case, the vaporizer is heated by a lamp at the start, but after a few minutes of operation the walls of the vaporizer, if well protected, remain at a dull red heat, due to the heat of compression and explosion.

Capitaine * employs a method which differs from that used in the Hornsby engine in that, at the moment the fuel is injected into the vaporizer, a little auxiliary air is also admitted which sweeps the oil vapor formed into the combustion chamber proper, where it meets the main body of air and is compressed with it. Ignition ensues from the hot walls of the vaporizer as in the other cases.

4. **Ignition by Means of the Electric Spark.** — Electric Ignition is to-day used more than any other. In fact in some branches of the industry, automobile work for instance, it is used exclusively. The reasons for this are not far to seek. As compared with the hot tube, there is no flame, and no fuel required to feed it. The system is perfectly flexible and susceptible of perfect timing.

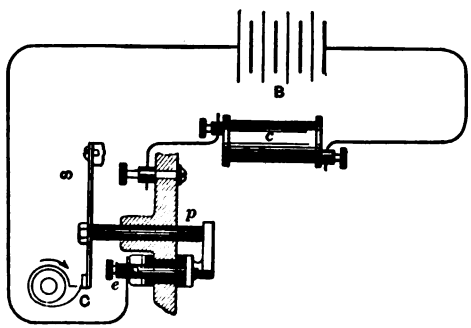


FIG. 13-6.— Make-and-Break Circuit.

There are a number of electric-ignition systems in use, differing in their methods of wiring, their sources of current, etc., but considering for the moment nothing but basic principles, all the systems may be grouped under two heads. These are:

1. Make-and-Break Ignition, and
2. Jump-spark Ignition.

In what follows, only the elementary principles of electric ignition will be discussed. For a comprehensive exposition of the subject, consult "Electric Ignition for Motor Vehicles" by W. Hibbert.†

1. MAKE-AND-BREAK IGNITION.—The simplest kind of make-and-break circuit is shown in diagram in Fig. 13-6.‡ In this figure, *B* is a source of current and *c* a so-called spark coil. In this case such a coil consists merely of a number of turns of comparatively heavy wire wound about a bundle of wrought-

* Zeitschrift d. V. d. I., 1907, p. 919.
† Whittaker & Co., 64-66 Fifth Ave., New York City.
‡ Roberts, The Gas Engine Handbook.

iron wires. This coil is in series with the circuit, and acts as an inductive resistance. When the circuit is broken, it serves to intensify the pressure, causing a hot spark at the point of break. For this reason the writer prefers to call this kind of coil an intensifying coil rather than a spark coil, which, as used in jump-spark ignition, is a very different thing.

The make-and-break mechanism consists in this case of a stationary electrode, *e*, insulated from the rest of the engine, and a movable electrode, *p*. The latter is connected to a flat spring, *S*, which in turn is in contact with the cam, *C*. The current flows from *B*, through the intensifying coil, *c*, to the electrode, *p*, and from here through the electrode, *e*, back to *B*, thus completing the circuit. The operation is as follows: Cam *C*, rotating in the direction of the arrow, first presses electrode *p* against electrode *e*, making the circuit. At the proper moment, spring *S* slips off the cam, suddenly forming a gap between *p* and *e*, across which the spark jumps.

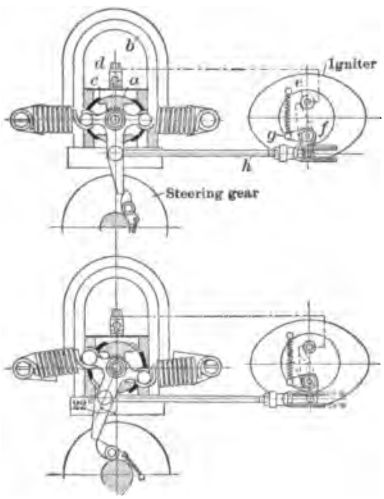


Fig. 13-7.—Make-and-Break Ignition Apparatus.

This type of make and break mechanism is known as the hammer break. An example from practice is shown in Fig. 13-7.* The source of current in this case is a Bosch magneto. The current flows from *d* to *e*, the stationary electrode, and returns through *f*, the movable electrode, and the forked rod, *h*. Actuated by a latching arrangement on the half-time shaft, the armature lever is pulled to the left about 20 degrees, as shown in the lower figure. This puts the two powerful helical springs shown in tension, so that when the latch releases, the armature sleeve instantaneously returns to its normal position, generating the required current by cutting the lines of force with great rapidity. At the same instant the fork, *h*, strikes the bell crank, *g*,

* Güldner, Verbrennungsmotoren, p. 365.

thus separating *f* from *e*, and causing the spark. This method is susceptible of adjustment by regulating the time of release of the latch.

It should be noted that in this particular instance the two electrodes are in contact until the spark is desired. This is admissible because with the source of current used, electric energy is generated only for an instant, just before the break. If a continuous source of current, such as a battery for instance, is used, it becomes necessary to modify the mechanism so that the electrodes are in contact only for a short time before the break; otherwise the system would be very wasteful of current.

The hammer-break mechanism is open to two objections, rapid wearing away of the points and fouling. The former is aggravated if too strong a current is used. It is usual, therefore, to make the points of contact of some metal that will not easily corrode or wear away under heat. Platinum, or platinum-iridium is extensively used for this purpose. There are, however, some special alloys on the market, such as Baker & Co.'s "Special," which are somewhat less costly, but do the work fully as well or better. Platinum is practically indestructible by heat, but it is hardly hard enough to stand the wear. The only remedy for fouling is periodic cleaning, although the claim is made for some of the special alloys that they remain bright indefinitely.

To overcome the objection of fouling, a modification of the make-and-break system known as the wipe spark is sometimes employed. In this, one electrode is made to revolve and a projection on it "wipes" across the other electrode at the proper time, causing a spark on the break. The spark produced in this way is perhaps hotter than that formed by the hammer break, and fouling of the sparking surfaces is effectually prevented. On the other hand, the wear is much greater.

Make-and-break ignition has the advantage that only a low voltage is required to operate it. The pressure ordinarily used is from six to eight volts, while in many cases from two to four volts is quite sufficient. There is thus much less danger from leakage of current and short-circuiting in this system than there is in the jump spark method.

The disadvantages of the system regarding wear and fouling have been already pointed out. Another, as compared with the

jump spark, consists in the fact that some mechanically operated gearing is required to "trip" the igniter. Both this fact and the rapid wear of the contact points have led designers to adopt the jump-spark system for high speed work. Lately, however, there

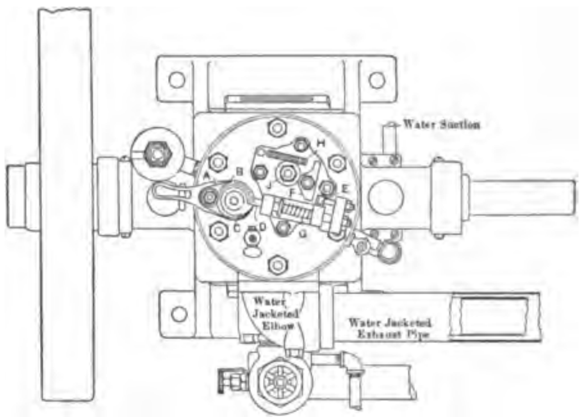


FIG. 13-8.— Plan, Fay & Bowen Engine.

seems to be a tendency to adapt the make-and-break system also to high speeds, caused no doubt by the very obvious disadvantages of the jump spark. There are ways of efficiently operating a high-speed trip gear. One of these, used by the Fay & Bowen Engine Co. for medium high speeds, is shown in Figs. 13-8 and 13-9. Fig. 13-8 shows a plan view of the vertical engine. The igniter shaft, *A*, passes vertically downward through the water-jacket space, and is driven from the crank shaft by a pair of bevel gears. It runs in bronze bearings, and where it passes through the jacket it is encased in a water-tight tube. At the top this shaft carries a small gear which meshes with a somewhat larger gear, *B*, under the igniter cam, *C*. All the gears are cut gears and run practically without noise. The driving is positive and

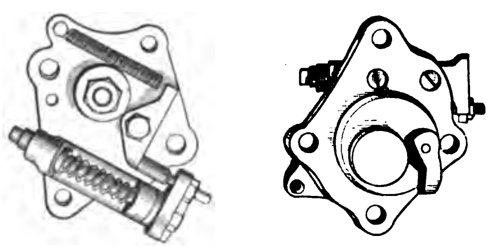


FIG. 13-9.—Igniter Block, Fay & Bowen Engine.

there can be no slip. As the cam, *C*, revolves, it engages the plunger *D* and forces it back against the spring *G*. When the plunger slips off the cam, the hammer, *E*, strikes the movable electrode, *F*, separating it from the stationary electrode, *J*, causing the spark. This action is carried out with the same rapidity no matter how fast the fly-wheel is revolved at the start. For the greater part of its revolution, cam *C* is not in contact with the plunger, *D*, and hence the electrodes are separated, thus maintaining an open circuit for the greater part of the time and preventing the waste of current. Just as soon as *C* commences to push back the plunger *D*, the spring *H* pulls back the movable electrode *F*, and contact is made for a sufficient length of time to insure a good flow of current.

The igniter plug, shown in greater detail in Fig. 13–9, is entirely independent of the driving gear and is held in place by four bolts, which can be removed at a moment's notice. The seat of the plug is a ground joint. The spark points can therefore be examined and the plug replaced in a very short time, or a new plug may be substituted for the old one. The chances for wear in the whole arrangement, however, are very small and there seems to be no reason why this igniter gear should not be used for speeds much higher than those for which the designers now use it. Adjustment of the spark in this gear is made in a very simple way by pivoting the gear *B* about the center of the shaft *A*, thus changing the position of the cam, *C*, with relation to the plunger. The adjustment is controlled by the hand lever shown. Should the lever by any accident be left in the advanced spark position, so that the engine may get an explosion turning it the wrong way the next time it is started, a small clutch located under the igniter cam, *C*, immediately frees the cam so that no second back explosion can take place.

2. Jump Spark Ignition.—Figure 13–10* shows diagrammatically the simplest type of jump-spark system. There are in all cases a primary or low-tension and a secondary or high-tension circuit. The primary circuit is shown in heavy line and contains the source of current, *B*. The current flows from *B* through an arrangement, *T*, called the interrupter, commutator, or timer, which serves to make and break the primary current at

* T. H. White, Petrol Motors and Motor Cars.

the proper time. It then passes through the primary winding, *P*, of the spark coil and returns to the source, completing the primary circuit. The secondary circuit, shown by a light line, consists of the secondary winding, *S*, of the spark coil and a spark plug in the cylinder of the engine, indicated in the figure by *Z*. It should be noted, in connection with the secondary circuit, that this circuit is never actually closed, since a spark gap always exists in the spark plug. Hence current cannot be said to flow in this circuit until the tension or voltage becomes high enough to bridge this gap by a spark.

To understand the operation of the jump-spark system it is necessary first to study the action of the spark coil. There are two kinds of these coils, the non-trembler and the trembler coil.

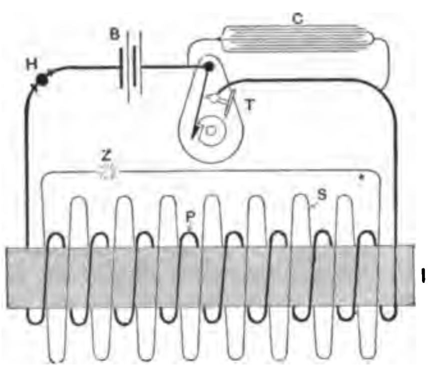


FIG. 13–10.—Simple Jump-spark System.

The former is the type indicated in Fig. 13–10. Its actual construction is about as follows: *I* is the core of the coil consisting of a bundle of fine iron wires. This is covered with a layer of some insulating material, and around this is wound the primary winding. This consists generally of several layers of insulated copper wire, about No. 20 or 22. A light layer of insulation next separates this from the secondary winding, which consists generally of some 10 to 15000 turns of very fine insulated wire. Each layer of this wire is separated from the next by a layer of insulation to prevent short-circuiting under the very high pressures occurring.

To understand easily what follows, it is necessary merely to remember that if any conductor of electricity is moved across a magnetic field, or if a magnetic field is moved across a conductor, an electric current will immediately be set up in this conductor. Further, that if a current be passed through a conductor, a magnetic field will immediately be set up around the conductor.

Now, in the spark coil described, a current is sent through the

primary winding as soon as the contact is closed at *T*. This converts the iron core of the coil into an electro-magnet, setting up a strong magnetic field. The magnetic lines move outward across the windings of the secondary circuit and induce a high tension in this circuit. But, owing to self-induction, the building up of the magnetic field is much slower than the collapse of the field when the primary current is suddenly interrupted at *T*. The magnetic lines then move inward across the secondary winding with much greater rapidity, hence the pressure induced is much higher than that existing during the building up of the field, and, if the spark plug is right for the coil, a spark will bridge across the gap in the cylinder, igniting the charge. The fact that

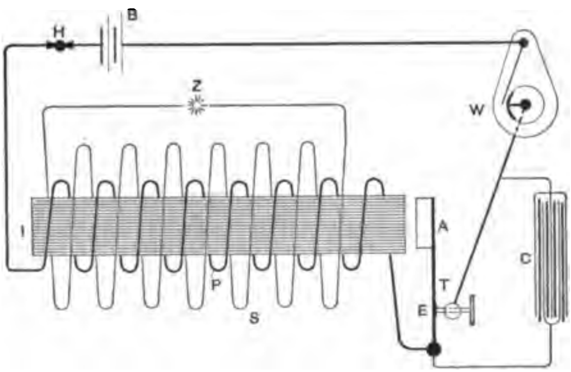


FIG. 13-11. — Jump-spark System with Trembler Coil.

the voltage induced on the making of the current is not high enough to bridge the gap prevents the occurrence of a double spark in the cylinder, which might lead to pre-ignition of the charge.

Since with the non-trembler coil the current in the primary is established only once when ignition is desired, only a single spark will occur in the cylinder. It is possible that this single spark may fail to fire and a series of sparks at the time of ignition is hence an advantage. This has led to the adoption of the trembler coil, Fig. 13-11*. The circuit shown in this figure is the same as that of Fig. 13-10, except that a trembler or "buzzer" *T* and a timer or commutator *W* have been substituted for the

* T. H. White, Petrol Motor and Motor Cars.

simple make-and-break mechanism, T, of Fig. 13–10. The action of the trembler is simple. As soon as W makes contact, the primary current converts the core, I, into an electro-magnet which attracts the armature, A, of the trembler blade. This action, however, breaks the primary current by pulling the spring blade, T, away from the constant screw at E. A spark then jumps over in the cylinder as before explained. The breaking of the primary current, however, releases the armature, A, which returns to its normal position, again establishing the primary circuit at E. The operation is then repeated. This action establishes a pulsating pressure in the secondary winding, causing a series of sparks as long as contact is maintained at W.

The advantage of the trembler coil has already been pointed out. The disadvantages exist in the fact that a second moving part is introduced into the primary circuit which must be kept carefully adjusted if the system is to work satisfactorily.

In both Figs. 13–10 and 13–11 it will be noticed that there is an arrangement, C, called a condenser, connected across the make-and-break mechanism, in the non-trembler coil across the interrupter, in the trembler coil across the vibrator or buzzer. The condenser consists of a large number of sheets of tinfoil, the number depending upon the capacity desired. Each sheet is separated from the next by a layer of insulation, and the alternate sheets are connected together. This manner of construction is clearly shown in the diagram. The object of the condenser is to prevent serious sparking at the make-and-break contacts in the primary circuit. The reasons why such a spark occurs at all in such a circuit is that the collapse of the magnetic field not only induces a high pressure in the secondary winding, but also causes a momentary increase in the pressure in the primary, thus bridging any small gap by a spark, and causing rapid wear of the contact points of the trembler at E, Fig. 13–11. When the primary circuit is now broken at E, the current induced, instead of jumping across, is expended in charging the condenser. The action is very similar to that of an air chamber on a hydraulic pipe line, absorbing shock by compressing air. The next time the primary circuit is closed at E, the condenser discharges and helps to send a current through the primary.

As actually constructed, spark coils are very compact. The

condenser is generally placed under the coils, and the whole is enclosed in a tight wooden box. Externally nothing shows but the terminals and the trembler, if the coil is of that type. In

Fig. 13–12. — Three-Terminal Spark Coil.

some coils one end of the secondary coil is connected to one end of the primary winding, so that only three terminals show, as in Fig. 13–12. Fig. 13–13 shows a four-terminal coil, the secondary terminals being on top. The tremblers are of various constructions, nearly each maker having his own design. They must give a quick break. They should be easy of fine adjustment, but the adjustment, once made, should stay. In many cases, as for automobile and marine purposes, the entire spark coil is enclosed in a second box with tight cover, so as to prevent fouling by mud or water. An example of this is shown in Fig. 13–14.

Fig. 13–13. — Four-Terminal Coil.

Timers. — A very important part of a jump-spark system is the device making and breaking the primary circuit, for everything depends upon the non-failing regularity of its performance.

There is a large number of such timers or commutators on the market, all more or less good. Figs. 13–15 to 13–18 show a few of the designs. The fundamental idea in all of these is the same. The half-time shaft actuates a cam or wiper inside of a case, which cam, at the proper time, makes and breaks contact with insulated terminals held by the surrounding case. The number of such terminals depends upon the number of cylinders. A great deal of ingenuity is shown in the prevention of friction between the cam and the terminal. The action of the Sintz timer, Fig. 13–15, is obvious. Here we have roller contact, the ends of the terminals are hardened steel, and the case is dust proof. Of somewhat similar design is the Lacoste timer, Fig. 13–16. The cross-section shows clearly the manner of construction. Somewhat more complicated, but of excellent design, is the Pittsfield timer, Fig. 13–17. In the Crouse-Hinds double ball timer, Fig. 13–18, the cam on the half-time shaft passes between two steel balls, held as shown. This makes the contact positive, keeps the surfaces clean, and the wear is very small.

Fig. 13–14. — Dash-board Coil.

With any of the above devices, the time of sparking may be varied by shifting the terminals with reference to the cam or wiper on the half-time shaft. Some timers incorporate governors to automatically time the spark.

Fig. 13–15. — Sintz Timer.

Spark Plugs. — A spark plug consists of two electrodes or sparking points which are held a certain distance apart in the cylinder. The central electrode is insulated, while the metallic

jacket enclosing the insulation generally carries the other spark point. This point, therefore, can be put in the circuit by fastening a wire anywhere to the engine. The essential requirements of

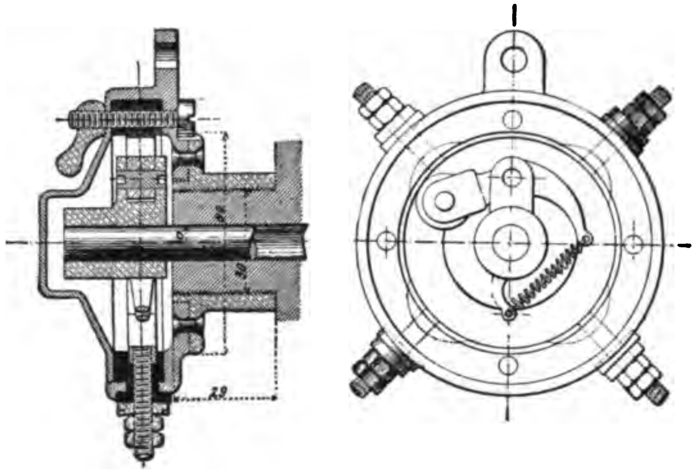


Fig. 13-16. — Lacoste Timer.

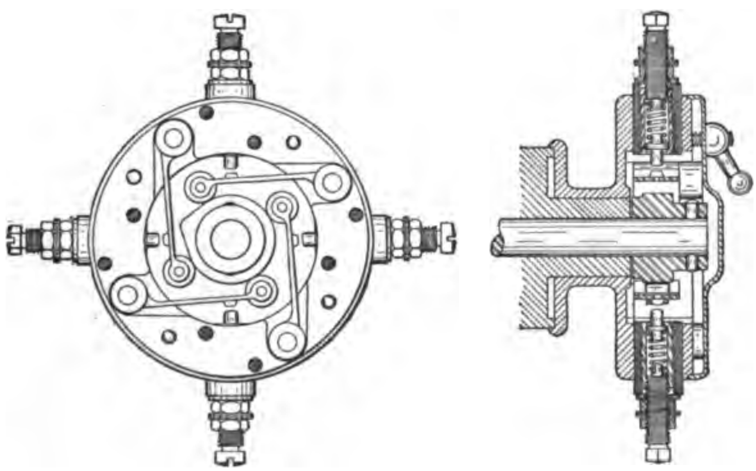


Fig. 13-17. — Pittsfield Timer.

the construction are that the insulation of the central electrode be sufficient and not liable to breaking down, and that the electrode points be so constructed that the plug is not easily subject to fouling.

The most important part of the entire plug is perhaps the insulation of the central electrode. Among the materials used for this purpose, porcelain and mica take the lead. Porcelain, while excellent, is very liable to break under any uneven expansion by heat, and the insulation must therefore be carefully designed with this point in view. Mica is not open to that objection and its electrical resistance is very high, but owing to its laminated structure, oil or soot may after a time be forced between the laminations under the high pressures existing, thus short-circuiting the plug.

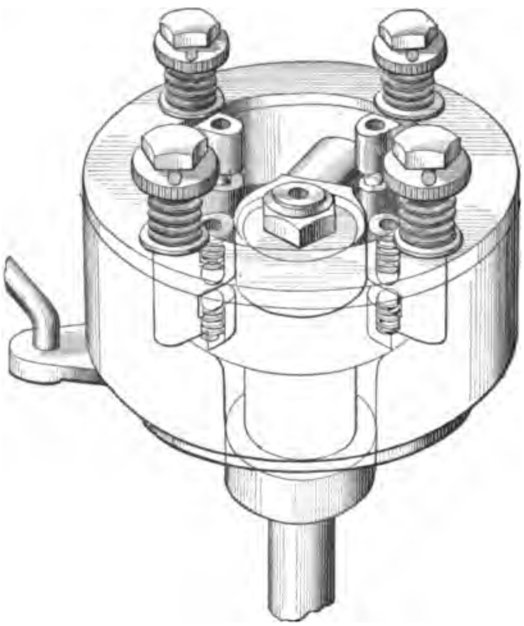


FIG. 13-18.—Crouse-Hinds Double Ball Timer.

How various manufacturers have tried to take into account the requirements mentioned, is shown in Fig. 13-19.* The first six plugs there shown have porcelain, and the last two mica insulation.

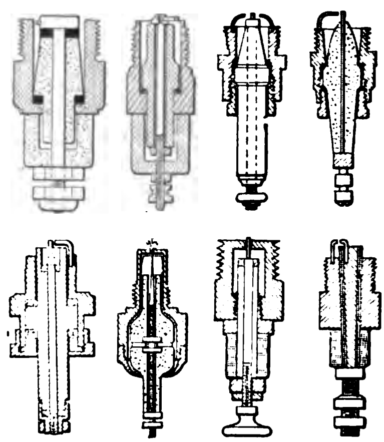


FIG. 13-19.—Various Designs of Spark Plugs.

It should be remembered in connection with spark plugs, that since the electrical resistance across the spark gap is greater when in actual operation in the engine than when in ordinary air, a plug may give a fair spark

* From Homans, Automobiles, p. 284.

when tested in air, and may still fail in operation. It should also be remembered that heat will lower the electrical resistance of porcelain, so that when the plug is very hot, short-circuiting through the insulation may result. It has been shown by experiment that while the resistance of porcelain cold was about 100 megohms, this fell to 2 megohms when the plug was at a dull red heat and under this condition sparking ceased. The spark, however, was immediately restored by an external spark gap, and continued even when the resistance had fallen to 800,000 ohms. Without the external gap, if the plug was allowed to cool down, sparking recommenced when the resistance of the porcelain had again risen to 5 megohms.

Auxiliary Spark Gap. — As the name implies, this is a second spark gap placed in the secondary circuit outside of the

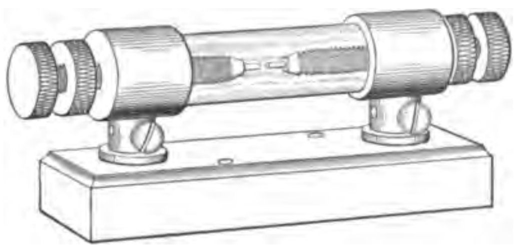


Fig. 13–20. — Auxiliary Spark-gap.

cylinder. This gap acts like an electrical condenser, above explained. The pressure builds up on one of the terminals of this gap, until it is high enough to break through the intervening air, causing an impulse of very high pressure through the circuit, thus giving a good spark across the main gap in the cylinder. Fig. 13–20.* shows one form of auxiliary spark gap. The advantages claimed for the device are:

(*a*) Greater certainty of sparking in the cylinder, since the higher pressure generated will cause a spark even across a partially fouled plug.

(*b*) Greater life of battery, since current cannot leap across a fouled plug as long as the auxiliary gap is not bridged.

(*c*) The sparking can be watched, since a spark across the gap always means a spark in the cylinder.

* Homans, Automobiles, p. 290.

In spite of these facts the auxiliary spark gap has not found extended application.

RELATIVE ADVANTAGES AND DISADVANTAGES OF MAKE-AND-BREAK AND JUMP-SPARK SYSTEMS

Make and Break Ignition. — Low tension throughout the circuit, requiring less thorough insulation, and causing less trouble from short-circuiting. The system is electrically more simple, while mechanically it is somewhat more complex than the jump-spark system. This latter fact makes it somewhat difficult to apply to high-speed engines.

Jump-spark Ignition. — Electrically more complex than the other, but has no moving parts inside of the cylinder.

Can be operated under very high-speeds with entire success, and has the greatest flexibility with regard to spark adjustment.

Sources of Current. — All sources of electrical current used for electric ignition may be classed under two heads:

1. Chemical Generators, under which come

(*a*) Primary sources, as wet and dry cells, and

(*b*) Secondary sources, as the storage battery or accumulator.

2. Mechanical Generators, variously called dynamos and magnetos.

1. *Chemical Sources of Current.*

(*a*) WET AND DRY CELLS. — All chemical cells consist of three essential parts, a positive and a negative electrode and an exciting liquid, called the electrolyte. As the name implies, in the wet cell this electrolyte is used in its liquid form, while in the dry cell it is mixed with some absorbing material, and the paste is used to fill the space between the electrodes. Take the dry cell as an example. The negative element is usually a carbon rod placed at the center of the circular case which forms the envelope of the cell. This rod is surrounded generally first by a layer of manganese dioxide, the purpose of which will appear later, and the rest of the space between this and the positive element, usually zinc in the shape of a cylinder, is then filled with the electrolyte paste, the original liquid being usually sal-ammoniac and water. The top of the cell is then covered with pitch or other substance that prevents the evaporation of the liquid in the paste, except that a small vent hole is left to allow

of the escape of any gas that may form within the cell due to the chemical action going on. In such a cell the current generated by the action of the electrolyte passes from the zinc to the carbon electrode, so that as far as the terminals of the cell are concerned, the carbon is the positive terminal. The chemical action destroys the zinc after a time and produces hydrogen gas on the carbon element. The greater the amount of this gas deposited on this element, the slower the generation of current, so that it may finally cease altogether. The cell is then said to be *polarized*. In dry cells the gas is taken care of in two ways; the vent hole in the top allows some of it to escape, while the layer of manganese dioxide above mentioned absorbs another part. But it is a fact that by these means not all of the gas is rendered harmless and hence the cells will polarize with more or less rapidity. This merely means that if current is drawn from them continuously for any considerable length of time, their strength will fail, making the cell appear dead. The same reasoning applies to wet cells where the hydrogen is allowed to escape through the liquid. Now, assuming that the zinc is not yet destroyed, if a cell so polarized is allowed to recuperate, it will again attain nearly its normal strength and may be used as before. The cells are said to be adapted to "open-circuit work." From all of this it is quite evident that in places where the requirement for current is not very great and, above all, not continuous, the primary cell will give satisfactory service. But where the draft of current is nearly continuous, as in high-speed four-cylinder machines for instance, the cell will rapidly polarize and soon fail to give sufficient voltage to operate the spark coil. The average size of a dry cell is about 2½″ x 7″. It will give when fresh from 1.3 to 1.5 volts and from 12 to 15 amperes.

It should be understood that there are other combinations of electrodes and electrolytes which may be used to generate current. Thus the so-called soda-cell is made up of a zinc plate and a copper-oxide plate with a caustic soda solution as the electrolyte.

(*b*) Storage Batteries or Accumulators. — A storage cell, like a primary cell, consists of two electrodes dipped in an electrolyte, but contrary to the primary cell, it cannot give off electrical energy in its original state when the circuit is closed. It is necessary to charge a storage cell before it can return electrical

energy on the discharge. The charging action causes chemical changes in the material of the electrodes and in the electrolyte. The energy so rendered latent is nearly all restored, when, after the charging current is disconnected, the outside circuit is closed. Chemical changes, producing a current in the reverse direction, then take place in the cell, which return both the electrodes and the electrolyte to their original condition. Some exhausted primary cells may be partially restored by passing a current through them in the reverse way, but in most cases the trouble is not worth while. The possibility of a nearly complete regeneration of a storage cell is the chief difference between it and a primary cell.

There are a number of materials which can be used as electrodes and electrolytes, but the usual type of storage cell to-day is that using some lead compound for the former and sulfüric acid and water for the latter. Hence only this lead storage cell will be here considered.

In its modern form, both the positive and negative plates of a cell consist of cast grids of lead, to which antimony is sometimes added to stiffen them. The perforations in the positive plate are first filled with some compound of lead, as Pb_3O_4, which is afterward converted to peroxide of lead, PbO_2. Similarly the negative plate is filled with PbO which is afterwards converted into spongy metallic lead. A number of plates so prepared are then placed side by side in a glass jar, or if the battery is to be used for automobile work, in a vessel of hard rubber or of wood lined with rubber or lead. Positive and negative plates alternate, and all the plates of like kind are connected together. There should always be one more negative than positive plates so that each side of each positive plate shall face a negative plate. The arrangement of plates presents a large plate surface in a compact space. Suitable insulation separates the plates from each other and keeps them from touching the bottom, in order to prevent any short-circuiting by contact or by dipping into any sediment that may form. In automobile batteries, the top is enclosed to prevent the spilling of the electrolyte, and nothing shows but the two terminals and an opening for filling. This is usually kept closed by a rubber cork with a small vent hole to allow of the escape of any gases that may form.

The chemical reactions that occur during charging and discharging are not yet fully understood, but it is agreed that the main action is the formation of lead peroxide on the positive and metallic lead on the negative plate during charging, and the formation of lead sulphate on both plates during discharge. The action is best explained by the following diagram.*

$$\xrightarrow{\text{Discharging}}$$

Charged Condition				Discharged Condition		
+ Plate,	Electrolyte,	− Plate,		+ Plate		− Plate	
PbO_2	$+ 2H_2SO_4$	$+ Pb$	$=$	$PbSO_4$	$+ 2H_2O$	$+ PbSO_4 +$ electric energy

$$\xleftarrow{\text{Charging}}$$

Such a lead cell when fully charged should show a voltage of from 2.2 to 2.25 on open circuit, and from 2.10 to 2.15 when the engine is running. The voltage soon drops to 2.0 and then slowly to 1.8. Three-quarters of the total discharge takes place between the latter figures. It is usual to discontinue discharging a cell when the voltage has reached 1.75. Beyond this point the formation of the insoluble lead sulfate becomes troublesome and, discharging much below this figure, the cell may be destroyed or at least seriously impaired.

Rating of Storage Batteries. — The amperage of storage cells depends on the weight of material in the cell converted by the chemical reactions, while the rate at which electrical energy can be taken off depends upon the surface of the active materials exposed to chemical action. Cells are rated by their ampere-hour capacity and nearly every maker states the normal rate of discharge recommended. For ordinary constructions the normal discharge rate is about .04 ampere per square inch of total positive plate surface, and the discharge capacity about 4 ampere-hours per pound of plate, including negative and positive. In order to be able to compare different cells, the capacity rating is based upon a current that will cause the voltage of the cell to fall to 1.75 volts in eight hours. Thus, if to produce this result a current of say 25 amperes must be drawn, the capacity of the cell is said to be 8 x 25 = 200 ampere-hours. If the rate of discharge is faster than this, it is obtained at the expense of capacity. Thus if a current of 40 amperes were drawn, the capacity might

* International Library of Technology.

be only 160 ampere-hours. Conversely, if the rate of discharge is slower than the standard, the limiting voltage of 1.75 may not be reached for say twelve hours instead of eight. These variations depend largely upon the make of cell.

Charging a Cell. — In charging a cell it is absolutely necessary to determine the polarity of the terminals of the source of current. The positive terminal must be connected to the positive terminal of the cell. The charging rate of lead cells should be about the same as the normal eight-hour discharge rate. It is, however, possible to use smaller currents for a longer time. The voltage of the charging current must be somewhat greater, from 5 to 10 per cent, than the discharge voltage, on account of the internal resistance that must be overcome. In one charging test, the charging voltage rose from 2.05 to 2.15 at the end of two hours, to 2.20 at the end of six hours, and to 2.50 volts in eight hours and forty-five minutes. The rate of charging was thus about normal. If charging is continued beyond this point, the electrolyte will have the appearance of boiling, owing to the gas that is being evolved. Slight overcharging will not injure a cell, but a large amount of it leads to sulfating and permanent injury.

Testing of Storage Batteries. — Two tests may be made, one for voltage, the other for sparking. For the former a low-reading voltmeter, 0 to 3 volts, is connected across the terminals of the battery, while the engine is in operation. The reading should be above 1.75 volts. Any cell may give 1.9 to 2 volts *on open circuit*, even if completely run down a short time before. The sparking test is made to determine in a way the state of the charge by noting the kind of spark. This test should be carefully done and not repeated too often. It is a dead short-circuit method and therefore not good for the cell. The use of an ammeter is for that reason not recommended, as it would take too long to get a reading. The sparking test is made by placing one skinned end of a piece of insulated copper wire in contact with one end binding post of the battery, and then drawing the other end rapidly across the other post. The spark should be loud and snappy.

Any storage battery should last from three to four years if properly treated. It is well to adopt a regular charging period,

say once in three weeks for the ordinary automobile battery, whether the battery is run down or not.

2. *Mechanical Forms of Generators: Dynamos and Magnetos.* — Mechanical forms of current producers have the advantage over primary and secondary batteries in that the energy required by them is derived directly from the engine they operate. Hence current will be produced as long as and only when desired. The other forms of generators depend upon sources of energy entirely extraneous to the engine plant, and the supply of current is therefore not in any sense automatic, which would be the ideal condition. The terms dynamo and magneto have been variously used. Some writers designate by "dynamo" any generator having electro-magnets serving to establish the magnetic field, and by "magneto" any machine employing permanent magnets for this service. Others define the difference as existing in the kind of current produced, a dynamo furnishing direct, *i.e.*, continuous current, while a magneto produces alternating, *i.e.*, pulsating, current. Whatever definition is adhered to, it should be remembered that in either machine the current is produced by an electrical conductor cutting the magnetic field. The current is produced in exactly the same way, and for exactly the same reason, as that established in the secondary winding of a spark coil, as explained above. In this case the conductor of electricity is wound upon a piece of metal, called an armature, which is rapidly rotated in a magnetic field. It makes no difference whether this field is produced by permanent magnets or by electro-magnets. If there are a number of such conductors upon the armature, and the current induced in each is properly collected by a so-called commutator upon the armature shaft so as to be practically continuous in its flow through the external circuit, we have what is generally called a dynamo. On the other hand, if the current in the external circuit rises to a maximum value and then dies out to give a maximum value next in the opposite direction, the machine is generally known as a magneto. While in all dynamos and most magnetos the armature constantly rotates in one direction, it should be stated that in all magnetos this is not at all necessary. Thus in the Simms Bosch magneto, the armature is stationary, and only a sleeve surrounding the armature is rapidly oscillated in the magnetic field. It would be beyond the scope

of this book, however, to discuss all the possible modifications, and the reader is hence referred to the works upon this subject.*

In general, the small dynamo used for ignition purposes is driven by means of a friction wheel from the fly-wheel of the engine. There is then no current available from the dynamo when the engine is started, and it becomes necessary to use a battery of some kind for the first minute or two, switching in the dynamo when it is up to speed. This scheme has the disadvantage that the battery is sometimes left in the circuit and the dynamos have been known to burn out under excessive engine speeds. A device called the Auto Sparker, Fig. 13–21, overcomes these difficulties. This little dynamo is fitted with a centrifugal governor which controls the position of the friction wheel on the fly-wheel rim, so that even at starting the armature rotates rapidly enough to furnish starting current. This does away with an auxiliary battery. As the engine speeds up, the governor of the dynamo acts to keep the armature speed constant, independent of the diameter of the fly-wheel or the engine speed. By adjusting the governor tension spring, it is possible to control the speed of the dynamo to get any current between 1 and 3 amperes and any voltage between 3 and 10 volts.

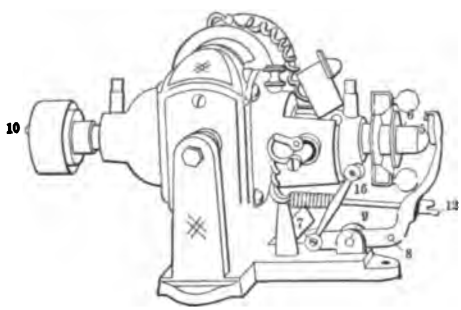


Fig. 13–21. — Auto Sparker.

Regarding magnetos, the following description of the action of a magneto, together with the explanation of the method of connecting it up, is taken from a catalogue of the Holley Bros. Company of Detroit. For clearness and simplicity this description can hardly be improved upon.

"A magneto, so far as its essential parts are concerned, is a very simple thing. It consists of a U-shaped piece of special steel, which is permanently magnetized; in other words, a common horseshoe magnet and a rotating armature. The armature consists of a soft iron core of approximate *H* cross-section as

* W. Hibbert, Electric Ignition for Motor Vehicles.

viewed along the shaft upon which it is supported and on which it is designed to rotate. The magnet, to the free ends of which are affixed soft iron arc-shaped pole pieces, and the armature core with the sides of the *H* correspondingly arc shaped, is shown in vertical section in Fig. 13–22. In the slot formed in the armaturo

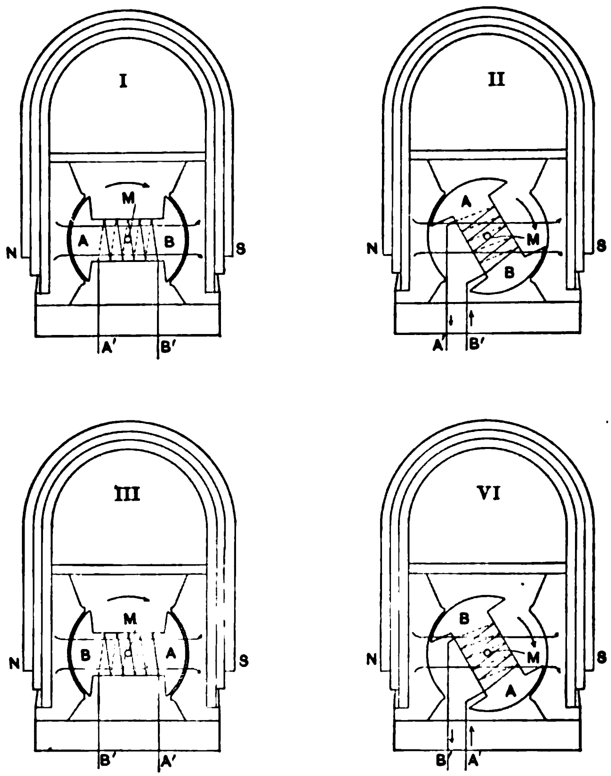


Fig. 13–22.

core by the sides of the *H*, wire is wound in turns lengthwise of the armature shaft. So much for the construction of the elementary magneto. In order to understand how it generates in its armature, when turned, an electric current, it is necessary to remember one law of physics, namely: Whenever a wire is wound about a magnetized soft iron core and the magnetism of the core suddenly dies out, there will be a tendency for a current to be

produced in the wire. A familiar example of the working of this law is found in the operation of the common jump-spark coil. Here we have a core made of soft iron wires and around it is wound a great many turns of fine wire, the ends of which are connected to a spark plug. The core is also wound with a coil of wire which is supplied with current from a battery, and when this current is flowing the core is magnetized. When the current from the battery is interrupted, the magnetism in the core suddenly dies out, and, in accordance with the law above stated, a tendency is created for a current to flow in the fine wire coil which is connected to the spark plug and this 'induced' current jumps at the plug.

"In order to explain how the iron core of the magneto armature with its winding is magnetized and how the magnetism of the core is caused suddenly to die out, it is necessary to refer to four diagrams of Fig. 13–22, showing the armature in different positions of rotation with respect to the pole pieces. In diagram (I) the armature is represented with the two heads of its core in close proximity to the faces of the pole pieces. The space between the pole pieces is thus almost completely filled or bridged with iron, and magnetism passes from one pole piece to the other through the armature core, thoroughly magnetizing it. Next consider diagram (II). Here the armature is shown rotated into such a position that one edge of each pole of the armature core is just leaving the vicinity of one of the pole pieces. As soon as this position is passed, the space from pole piece to pole piece is no longer filled with iron, but with air which is not a conductor of electricity. Thus very little magnetism passes from one pole piece to the other and the core is no longer traversed by the magnetic influence and suddenly ceases to be magnetic. This is exactly the condition prescribed by the above quoted law for the production of a current, and, in fact, when the armature in its rotation leaves position (II), there is a sudden impulse of current produced in the wire of the armature which dies away after the armature rotates a little beyond this position. In position (III), the conditions of armature magnetization existing in position (I) are reproduced, except that the armature has changed ends in respect to the pole pieces and the magnetic influence passes through it in the opposite sense, charging it oppo-

sitely, so that when the magnetism is discharged in position (IV) the current will be in the opposite direction through the wire of the armature winding. As the armature is turned upon its shaft, there are thus produced, in each complete rotation, two rather short impulses of current of opposite direction nearly corresponding with the instants at which the armature heads, so to speak, 'part company' with the pole pieces and are half a revolution apart. During the remainder of the rotation there is no current flowing. It may be readily seen that by connecting one end of the armature wire to the armature core, and by connecting the other to an insulated metallic contact segment, carried by the armature shaft, upon which bears a stationary insulated brush, the current impulses may be taken from the magneto for use.

"Now as to the practical use of such a magneto for ignition purposes. Since it is only during a small part of the armature rotation that current is being generated, it is necessary to rotate the armature shaft at such a speed that these electrical impulses shall be so timed as to correspond with the periods when ignition is required by some one cylinder of the engine. If this were not attended to, the ignition periods of the engine might occur during the parts of the armature revolution, when no current was being produced. In order to bring about this result, the magneto and the engine must, at all times, run at a properly proportioned ratio of speeds and the positions of the engine, crank shaft, and armature must be adjusted right in the first place. If the magneto shaft is geared to the engine at the right ratio and the teeth of the two gears are correctly meshed, the desired result will be brought about. For instance, if the engine be of the four-cylinder, four-cycle type, four sparks will be required for each two crank-shaft rotations. Four sparks will be produced for each two revolutions of the magneto, as well, and thus, if the magneto and the engine run at the same speed, the sparks will be numerically correct. If geared to the crank shaft, the crank-shaft gear and the magneto gear would have the same number of teeth, and if driven from a two to one shaft, the number of teeth in the two to one shaft gear would be twice as great as the teeth of the magneto gear. By changing the particular teeth of one gear which are in mesh with certain teeth

of the other, the current impulses may be made to occur at the moments when the pistons are exactly in the firing positions."

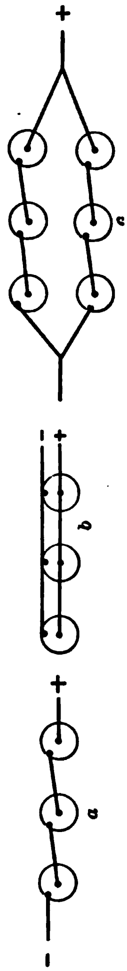


FIG. 13-23. — Methods of Connecting Cells.

In variable-speed engines, as automobile machines, for instance, the service required of the ignition outfit becomes more exacting as the speed increases, owing to greater compression and less available time. This in the case of mechanical current generators is met by a natural increase in voltage with increase in speed, which constitutes another advantage of this type of generator as compared with primary and secondary cells. Thus less hand manipulation of the spark is required, but all magneto systems should be provided with means of altering the armature position relative to the crank-shaft position in order to alter the time of spark.

Methods of Connecting up Primary and Secondary Batteries, and Systems of Wiring used. — Primary and secondary cells may be connected in series, in parallel (or multiple) and in multiple-series. The meaning of these terms is explained in Fig. 13-23. For series connections, Fig. 13-23*a*, each positive element of one cell is connected to the negative element of the next, leaving free the negative element say of the first cell and the positive element of the last for connection to the outside circuit. In the second or multiple method of connection, Fig. 13-23*b*, all the like elements of the cell are connected together. Fig. 13-23*c* finally shows six cells in multiple series, *i.e.*, three each are connected in series, and these two sets in multiple or parallel. To compute the voltage and amperage that each one of these combinations will furnish to the outside circuit, let

N = number of cells in the combination
V = voltage of one cell, and
A = amperage of one cell.

Then the following formulæ will give the desired information:

Kind of Combination	Series	Multiple	Multiple-series
Voltage of set...........	NV	V	NV
Amperage of set.........	A	NA	2 A

The ordinary dry cell, as stated, furnishes about 1.5 volts and 12 to 15 amperes. A *make-and-break* circuit should operate properly on about 8 to 10 volts, hence from 5 to 7 dry cells in series are required for this service. As far as *jump-spark systems* are concerned the following table gives pressures and currents required to operate some of the well-known spark coils, together with other interesting information.* From this table it is clear that from 4 to 6 dry cells in series are sufficient to operate most jump spark coils:

	Volts	Amps.	Vibration per sec. of Trembler	Prim. Res. Ohms.	Sec. Res. Ohms.
Kingston................			89		
Apple..................	5.20	2.2	94	.171	3715
Guenet	3.7	1.31	111	.300	2337
Guenet	5.8	....	123	....	2337
Hardy..................	3.78	1.05	122	.274	2779
Fisher..................	5.8	.82	149	.613	2590
Dow....................	3.84	.57	149	.210	5394
Lacoste.................	3.72	1.46	177	.232	2006
Lacoste.................	5.62	1.94	197	.232	2006
Heinze..................	3.66	1.31	210	.320	1302
Pittsfield................	....	....	228	....	....
Induction Coil Co........	3 62	1.55	360	.312	6180
Milwaukee	....	....	390	.312	6180

Turning next to the *systems of wiring used*, all make-and-break systems are *low-tension* systems, *i.e.*, the voltage does not generally exceed 8 to 10 volts. Fig. 13–24 † shows such a system in diagram with a magneto as the source of current. The circuit is easy to trace. One side of the electrical conductor on the armature is grounded, that is, connected to the engine frame

* H. G. Chatain in the *Automobile*, July 18, 1907.

† The following three figures are from an article by C. B. Hayward in the *Automobile*, April 4, 1907.

through the armature shaft and the frame of the magneto itself, as shown at G_1. The other end sends its current to one electrode

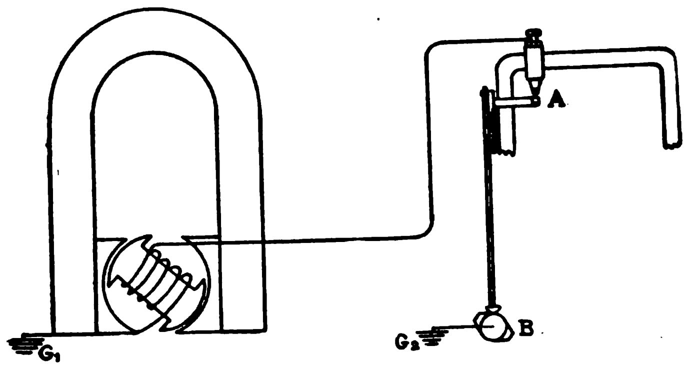


FIG. 13–24. — Simplicity of the Wiring of Low-tension Systems.

of the make-and-break mechanism at A. When the commutator or timer, B, makes contact, current flows, the circuit being completed by grounding B, as shown at G_2.

Jump-spark systems are called high-tension systems, but a distinction should be made depending upon whether *high or low*

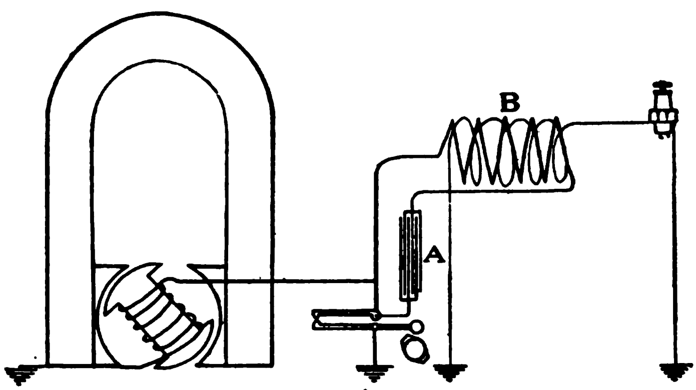


FIG. 13–25. — Wiring Diagram, "High-tension with Coil System."

tension magnetos are used. With a low-tension magneto it becomes necessary to use the ordinary spark coil, and hence this method is sometimes called the *high-tension with coil* system. Fig. 13–25 shows the wiring for such a system, A being the con-

denser and B the primary and secondary windings of the spark coil. It is comparatively easy to trace out the complete primary and secondary circuits, if one takes into account the proper ground returns.

The true high-tension jump-spark system differs from the above in the fact that the high-tension magneto embodies the secondary winding and the condenser of the spark coil. Hence the use of a separate spark coil is avoided. In Fig. 13–26 the two windings are indicated on the armature, but the condenser is shown at one side for the sake of clearness. This diagram shows the wiring for four plugs.

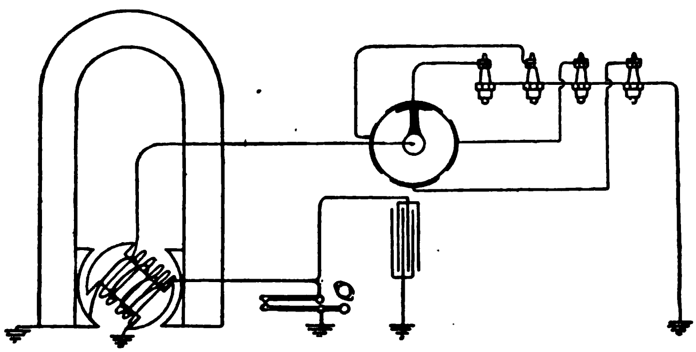


FIG. 13–26. — Wiring Diagram of True High-tension System.

HIGH-TENSION DISTRIBUTOR. — With the ordinary system of jump-spark ignition, as many coils as there are cylinders are required. It is possible, however, by placing a distributor in the high-tension side to serve a number of cylinders from one spark coil. The advantage of such a system is obvious, although it is bought at the cost of placing a make-and-break mechanism under very high voltage. The difficulties inherent in this, however, have been fairly successfully overcome. The difference in the wiring is made clear by Fig. 13–27 and Fig. 13–28,* both applying to four-cylinder engines. The former shows the four-part timer connected to the four spark coils serving the plugs $S\,P_1$ to $S\,P_4$. In Fig. 13–28, a high-tension distributor D connects the high-tension side of the single coil first with one plug, then with another as may be required. In practice the high-tension distrib-

* Both from Hibbert, Electric Ignition for Motor Vehicles.

utor *D* and the primary commutator or timer, *C M*, are combined in one device. Fig. 13–29 shows the Crouse-Hinds Double Ball Contact distributor and Fig. 13–30 the Leavitt distributor.

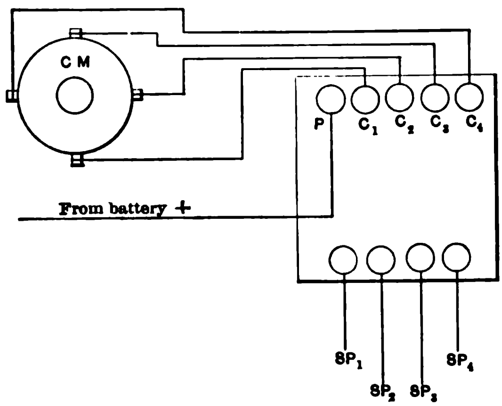


FIG. 13–27.

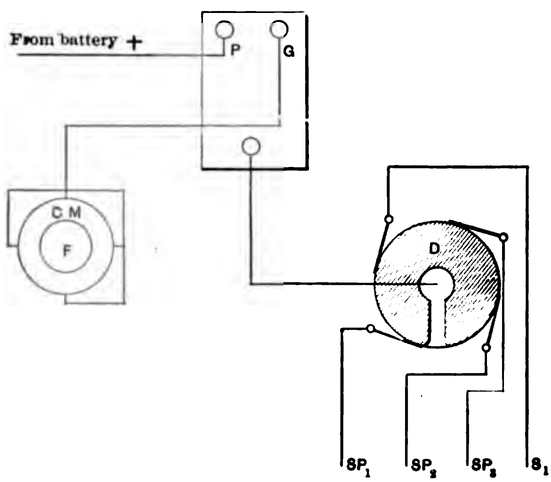


FIG. 13–28.

The essential thing in high-tension distributors is that serious sparking in the high-tension side must be avoided. For that reason, in most of the devices the primary commutator does not establish the current until the high-tension distributor is in con-

tact with its proper plug segment and the primary current is broken before the contact in the high-tension side is over. To quote from the description of the Crouse-Hinds device:

"The principle is exactly the same as that of the commutator already described, Fig. 13–18. The distributor has two cams and two sets of ball contacts, one set for the timer and the other for the distributor, the only difference being that in the latter the balls in each contact are about three-eighths of an inch apart and the cam insulated from the shaft. The connection is

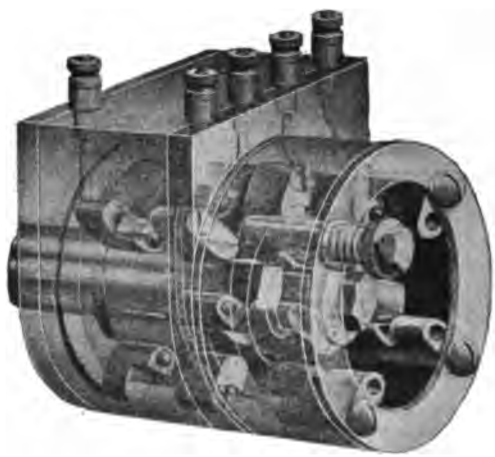


FIG. 13–29. — Crouse-Hinds Distributor.

made and the circuit is closed for each cylinder as the cam passes between the balls."

The following is a description of the Leavitt distributor as given in a catalogue of the Uncas Specialty Company:

"This device, two views of which are shown above, consists of a cylindrical casing, *A*, of hard rubber, into which is let a metal plate, *B*, at one end, and which is covered by a hard rubber cap, *C*, at the other end. Upon a ball bearing in the end plate, *B*, is mounted a driving sleeve, *D*, designed to be secured upon an extension of the cam shaft, and carrying fast upon it contact blocks, *E*, *E*, *E*, *E*, which make contact, successively, with the primary ball contact terminal, *F*. Thus far the device is identical with the ordinary timer. The commutator portion is located

at the opposite end of the cylindrical casing. The latter is enlarged at that end, and into the radial wall between the two cylindrical portions are clamped four flat-head studs, *G*, *G*, which serve as binding posts for the spark plug connections. Into the end of the metallic sleeve, *D*, is fastened a hard rubber stud, *H*, which at its outer end carries a radial arm, *I*, which is of metal with a relatively wide contact shoe at its end, which when the sleeve, *D*, revolves, passes over the four contact studs, *G*, thus conducting the current successively to the four plugs. The current is conducted to the rotating arm by a central ball contact, *J*, secured into the cap, *C*. When the hard rubber casing is moved around its axis by means of the arm, K, to vary the time of spark, both primary and secondary contacts are equally displaced."

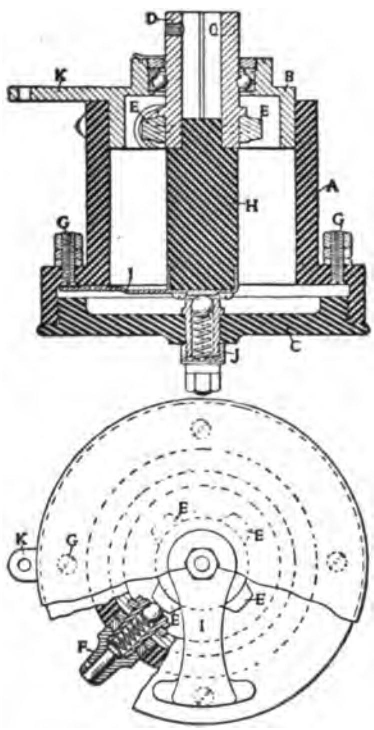


FIG. 13-30. — Leavitt Distributor.

Mufflers. — A muffler is an essential part of a gas-engine installation if quiet operation is desired. The sudden release of a body of gas at a pressure normally of 40 pounds per square inch above the atmosphere causes a sharp noise very annoying in the long run. A muffler is merely an enlargement in the exhaust pipe to allow of gradual expansion of the escaping gases. Many different schemes are used. Thus in some cases the muffler is merely a cast-iron pot or vessel of suitable volume, in other cases the muffler is of more elaborate construction consisting of a vessel filled with baffles or partitions in various ways and intended to expand the gas gradually and to break up the sound waves. Besides efficiency as a dampener of noise there are two other points that should be kept in mind

with regard to mufflers, absence of any serious back pressure and durability.

The increase in back pressure caused by a muffler depends upon the volume of the muffler and upon the amount of choking caused by the baffles. The minimum volume of the muffler should be at least five times the cylinder volume, but for complete silencing twice this volume is none too much. Outside of the plain cast-iron muffler pot, it is probably safe to say that nearly all baffled mufflers increase the back pressure somewhat. This fact is conceded by most manufacturers in that they furnish a cut-out which is called into service when the engine is to be called upon for a hard pull. At least one manufacturer, however, claims the production of a slight vacuum between muffler and engine due to the ejector action of the muffler.

There is little doubt that cast iron is the best material to use for mufflers, as it is least attacked both by heat and the action of gases. This is especially true if a spray is used in the exhaust pipe for the purpose of cooling and condensing the exhaust gas. Many mufflers, however, for the sake of lightness and ease of manufacture, are made of galvanized sheet steel and give quite satisfactory service.

The noise of the air rushing into the inlet pipes of an engine is also sometimes very annoying and in some cases may cause undesirable vibrations of doors and windows, and even walls, of the building. In such a case it is usual to muffle also the inlet pipes, and one of the best ways to do this in small and medium sized engines is to take the air from the hollow sub-base.

In some very large engines the proper silencing of the intake and the exhaust becomes quite a serious problem, as the ordinary muffler would become very large. The expedient sometimes used is to draw the air through an underground masonry duct of ample size leading in from the outside of the building, and to discharge through a similar duct into a chamber from which the gases finally escape. A spray of water into the exhaust pipe of such engines, close to the exhaust valve, helps materially.

Figures 13–31 and 13–32 * show two types of muffler sometimes used. In the first the stream of gas is merely divided, in the second each division is furnished with an enlargement de-

* Mathot, Engineering Magazine, July, 1907.

signed to decrease the gas velocity still further. Fig. 13–33 shows the Powell muffler. The partial section shows how the gas is broken up into many fine streams by passing it through perforated plates. In the so-called ejector muffler, Figs. 13–34 and 13–35, made by the Motor & Mfg. Works of Geneva, N. Y., a part of the gas passes straight out through a central pipe. The rest is made to pass a series of perforated cones as shown. It is claimed that the central pipe acts as an ejector serving to draw the gas through the cones, thus eliminating back pressure and even creating a vacuum ahead of the muffler.

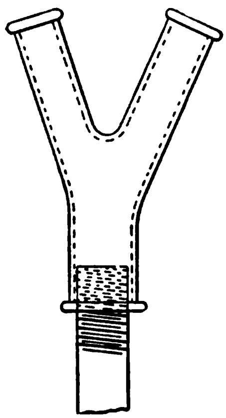


Fig. 13–31.

Starting Apparatus. — The best way of starting small engines, up to say 10 to 12 horse-power, is to turn the fly-wheel over by hand either in the direction of normal rotation until the engine picks up, or, after a charge has been drawn in, by turning it in the opposite direction against the compression and then snapping the spark by hand. In starting an engine in this way, it is essential to make sure first that the time of sparking is rather late, otherwise the engine may "buck," which may possibly lead to an accident to the person starting it.

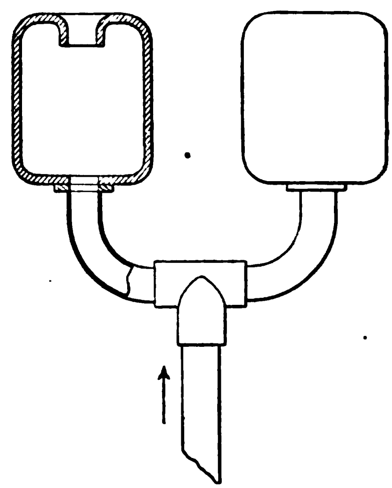


Fig. 13–32.

As the size of the engine increases, however, the manual labor involved in the above scheme soon becomes too great and other means had to be developed. These may be grouped under several heads. It should be noted that none of these are quite able to start an engine under load.

(*a*) Starting crank.

(*b*) Starting by smaller engine or other source of power.
(*c*) Starting by mixture.
(*d*) Starting by compressed air.
(*e*) Electrical starters.

Of the methods above mentioned, (*a*) is nearly universally used for engines up to 15 to 20 horse-power; beyond this, starting by compressed air, method (*d*), is generally employed.

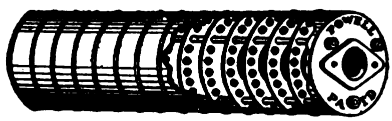


FIG. 13-33. — Powell Muffler.

FIG. 13-34. — Ejector Muffler.

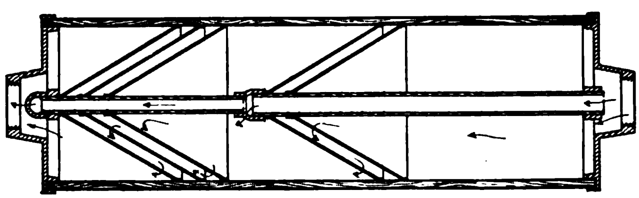


FIG. 13-35. — Ejector Muffler.

(*a*) Most starting cranks are so arranged that, when turned in the direction of rotation of the engine, they grip the shaft. As soon as the first explosions accelerate the shaft so that it turns faster than the crank is being turned, the latter is released. This scheme does not prevent the crank from "kicking" back into the starter's hand if the spark should happen to be early, and many accidents have resulted therefrom. A crank which avoids this drawback is shown in Fig. 13-36. The following description of this device is from the *Horseless Age*, March 7, 1906:

"A bushing *A*, having a thread of exceedingly high pitch cut on its interior surface, contains a threaded sleeve *B* which serves as bearing for the shaft of the starting crank *C*. Rigidly secured to the shaft of the crank *C* are a ratchet wheel *D* and a ratchet cam *E*, the latter adapted to engage with the ratchet *E′* on the motor shaft. In an extension of the sleeve *B* are a set of spring press pawls *F* which are adapted to engage with the ratchet wheel D.

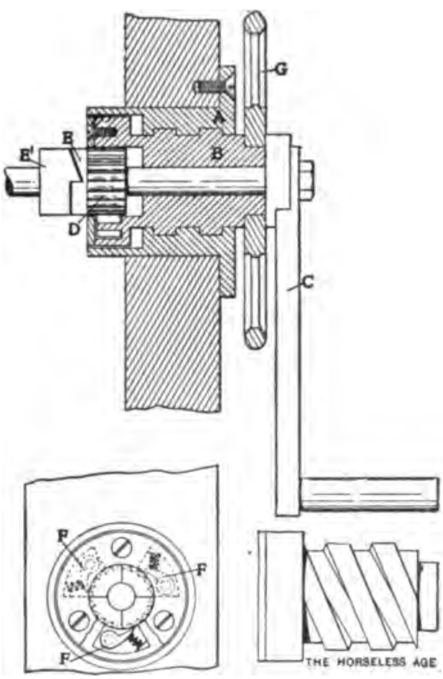


FIG. 13-36. — Starting Crank.

"The device is mounted at such a distance from the end of the crank shaft that the ratchet cams *E* and *E′* cannot engage unless the sleeve *B* is screwed to the limit of its motion into the bushing *A* by means of the hand wheel *G*. In starting the motor, the device being in the position shown in the assembly view, the action is the same as that of an ordinary starting crank. When the motor runs up to speed, owing to the pressure between the cam surfaces of ratchet cams *E* and *E′*, the starting spindle is forced back into the sleeve *B*, and the ratchet cams *E* and *E′* are thereby disengaged. However, if the motor should kick back, the pawls *F* would engage into the notches in the ratchet wheel *D*, and the sleeve *B* would be rotated and draw the ratchet cams *E* and *E′* out of mesh. The pitch of the thread on the sleeve *B* is so steep that a very slight rotation of the sleeve in the bushing will carry it back far enough to pull the starting spindle out of engagement with the crank shaft. The pawl and ratchet mechanism is completely enclosed by a lateral extension on

the bushing *A* and an end plate bolted to the extension of the sleeve *B*."

(*b*) If the plant has other engines in operation or if there is a shaft in operation it is quite easy to transmit the motion to the engine to be started. Some large engine installations were equipped with smaller engines the shaft of which carried a pinion which meshed with a ring of teeth on the fly-wheel of the large machine. When the large engine picked up its cycle, the small engine was automatically put out of mesh with the wheel. The cost of fitting a large engine for starting in this matter, however, is considerable. Hence this method has been largely replaced by the use of compressed air.

(*c*) Starting by means of the fuel mixture is done in various ways. The scheme appears to be reliable in the case of engines using illuminating gas; for other gases it never was in any extended use and in fact is to-day nearly obsolete. Besides the fact that if the first charge should fail there was generally not enough left for a second trial, the storing of an explosive mixture is obviously attended with some danger.

In nearly all cases of starting by the fuel mixture, the engine crank is put a few degrees above center, *i.e.*, well in the beginning of the expansion stroke. One method is to charge the compression space with air at atmosphere pressure, and then to open the gas cock. The mixture formed is allowed to escape through a special check valve and is ignited by a Bunsen burner just at the orifice. When the issuing flame burns reddish, showing that the mixture is rich, the gas valve leading into the cylinder is suddenly closed off. The flame at the orifice of the check valve strikes back into the combustion chamber and explodes the mixture remaining. The pressure generated closes the check valve and drives the piston forward. This is the original method of Clerk and of Green.

The pressure generated behind the piston in the above manner is not very high, and the velocity attained therefore correspondingly low. The next step in the development was then to compress the charge before ignition. The Clerk pressure starter and the Clerk-Lanchester high-pressure starter are of this type. Fig. 13-37 * shows the latter gear. The method of operation is as

* Clerk, The Gas and Oil Engine.

follows: As the engine is slowing down for a stop with the main gas valve closed off, valve *W* is opened. This draws fresh air into the chamber *D* and the pipe *D′* through the valve *Y*. When next the engine is to be started, the gas cock *F* is opened. Gas flows into *D* and *D′* and forms a combustible mixture. Through a cock on the cylinder, or a slightly open exhaust valve, a part of the mixture is allowed to fill the combustion chamber through the valve *W*, while another part escapes through a check valve *Y* and is ignited by the Bunsen burner *X*. As soon as the mixture is right, *F* is closed off, the flow stops, and the flame strikes in through *Y* into *D*. The pressure generated closes *Y*. The flame

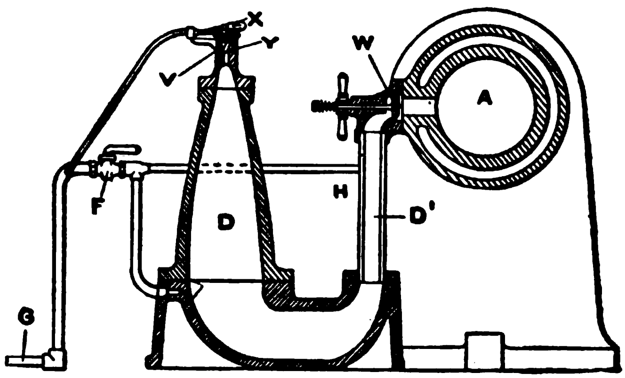


FIG. 13–37. — Clerk-Lanchester Starter.

and pressure wave travels around into *D′*, compressing the still unburned charge ahead of it and finally ignites the charge so compressed in the combustion chamber *A*. The starting pressure so obtained is about twice that obtained by the method first described.

Besides these flame starters, of which there are a number of modifications, other methods are in use. Thus in some engines a mixture is pumped under more or less pressure by hand into the combustion chamber, and ignited either by snapping the spark, or, as it is done by one manufacturer, by suddenly lighting an ordinary parlor match, the head of which is allowed to slightly project into the mixture. The device for doing this is quite simple but the method is not much used.

Another scheme is to charge a pressure tank with the combus-

tible mixture. This is generally done by means of a special valve which, when the engine is in operation, opens somewhere along the compression stroke of the engine, and allows it to compress some of its charge into the tank. The starting is then done either by merely filling the combustion chamber under low pressure from the tank and igniting the mixture by suitable means, or by opening the tank valve wide, utilizing the pressure of the compressed mixture to start the engine, igniting the charge on the next compression stroke in the regular way. This scheme has the advantage that if the first trial fails there is generally enough in the tank for several more attempts, but the storing of any considerable quantity of mixture under pressure is not to be recommended on account of danger of explosions in the tank.

Finally, at least one large German engine has been started by means of the mixture by first placing the crank in the proper position, and then blocking the fly-wheel by a plug of suitable material. A small gas engine is then started by hand and allowed to fill the combustion chamber of the larger engine under pressure in the same way as the tank was charged in the previous method. When the desired pressure is reached, the mixture is fired by hand manipulation of the spark, the impulse of the explosion breaks the plug holding the wheel and the engine picks up the cycle in the regular way.

(*d*) To-day the starting of internal combustion engines of any size is generally done by compressed air. The latter may be obtained in various ways. In some of the smaller engines, when the engine is to be shut down, the fuel valve is closed and the engine as it slows down is allowed to compress the air drawn in into the tank. Some other installations have a hand-operated air compressor for charging the tank, or the compressor may be belt-driven from the engine for a few minutes. It is hardly necessary to say that all tank and pipe connections must be absolutely air tight, because the failure of the air pressure in a plant of some size would cause serious delay. For engines of the largest capacity, it is usual to employ air compressors completely independent of the engine, thus obviating any failure due to tank leakage.

The method of starting by compressed air is to set the engine beyond the center and then to give an impulse through the starting valve. In some engines this valve is hand operated, in

others it is operated by the valve gear. Again in some, especially single-cylinder, engines the valve-actuating gear is not changed at all, except perhaps to relieve the compression somewhat, while in some multi-cylinder engines the gear on one cylinder is changed to make this cylinder act on the two-cycle principle, *i.e.*, so that it can take an air impulse every turn, until the other cylinders have started regular operation. In any case it is best to retard the spark somewhat for starting.

The starting pressures employed vary from 100 to 150 pounds,

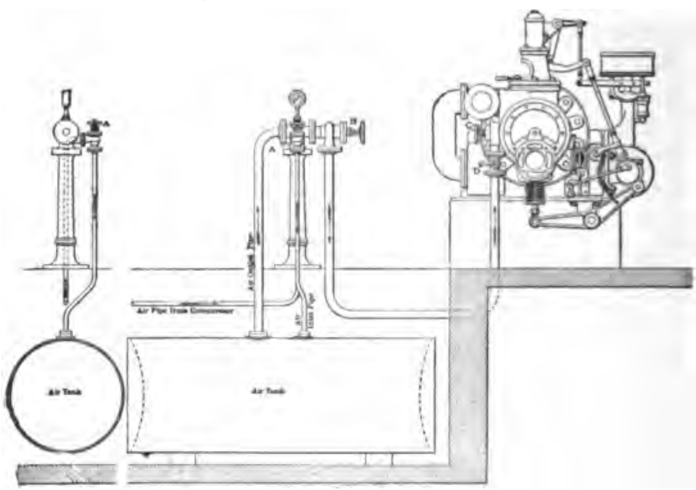


FIG. 13–38. — Air-starting Apparatus.

depending upon how the air is compressed. In general one or, at the most, two impulses are sufficient to start a machine.

The following description of Fig. 13–38, given by F. E. Junge in Power, April, 1906, shows the method of starting a large engine by compressed air:

"In this diagram *A* is the valve controlling the flow of air from a separately driven air compressor to the tank and *B* a similar valve in the pipe connecting the tank to the engine. Both valves are mounted on one pillar, which also has screwed on top of it a gage indicating continuously the pressure in the tank. Regulation of the supply of compressed air to the engine cylinder is effected by means of an automatic spring-loaded inlet poppet

valve, the stem and disc of which may be released or held fast by screwing down or unscrewing the hand wheel *C* at the engine end of the air pipe. A plug valve *D* is inserted in the air pipe immediately ahead of the inlet valve. Before starting the engine, the fly-wheel is turned into such a position that the crank is about 30 degrees above the inner dead center. The starting gear is adjusted so as to open both the inlet and the exhaust valves at the proper moments, the action being such as to allow part of the compressed air to escape during a fraction of the return travel of the piston and thereby reduce the compression to about 28 pounds per square inch for rich gases, and about 50 pounds per square inch for poor gases. In the meantime, the electric ignition device has been automatically adjusted so as to retard ignition for the first few strokes. The main fuel or gas valves must, of course, also be set so as to produce the most favorable mixture for starting conditions.

"To start the engine, the air stop valve *B* is opened, the automatic inlet valve released by screwing down the hand wheel *C* to the full extent, and compressed air is then admitted by turning the handle *D* 90 degrees. The piston will then begin to travel slowly on its outward stroke, and just before it reaches the outer dead center the handle *D* must be returned to its original position, shutting off the air supply. The first impulse given to the fly-wheel by compressed air will usually be sufficient to produce several revolutions at a speed of about one-fifth of the normal, when no load is on. During the following (suction) stroke a mixture of gas and air in the correct proportions is taken in, and on the next stroke compressed and ignited. If the right mixture does not happen to be obtained and ignition fails to occur, another compressed air impulse is given, which will always produce the desired result. After the first power stroke has been obtained the air supply valve *B* is closed and the automatic inlet valve held fast by unscrewing the hand wheel *C* until its hub bears against a collar on the valve stem, whereby the valve disc is firmly pressed on its seat. Then the starting gear is pushed back into the running position, so as to allow the mechanism to open the valves at the regular intervals only. The point of ignition is thereby automatically advanced and may now be adjusted by hand or by the governor of the engine so as to suit the changed conditions.

When the main gas admission valve is set in the correct running position, all operations for starting have been duly executed. It may be added that it takes less time to perform the complete cycle of operations than it takes to describe it."

(*e*) In central stations, or in other locations where electric current is available, starting by electricity is the simplest and most satisfactory method. There are again several ways in which this may be done.

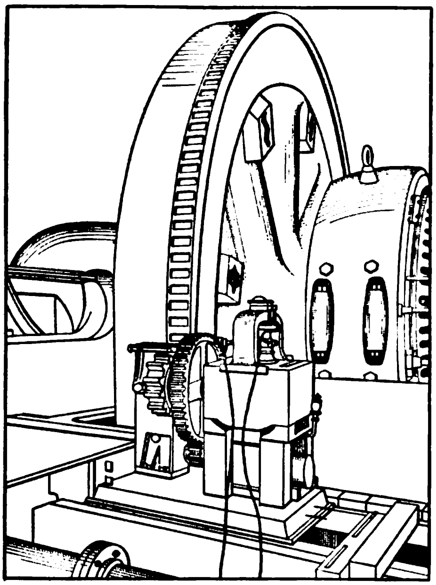


FIG. 13–39. — Motor Starter.

If the engine drives a dynamo, it is possible to drive this dynamo as a motor from another source of current, until the fly-wheel has attained sufficient velocity. The layout for doing this is not at all simple, however, and must be carefully handled. A storage battery charged during the previous operation may be used as a source of current if no independent source of current is available.

A simpler way is to gear an electric motor to a rack on the fly-wheel as shown in Fig. 13–39.* When the engine picks up the motor is automatically thrown out of gear. This scheme has the merit of great simplicity, absolutely no change in the valve operation of the engine being required for starting.

One method of automatically disengaging the motor is described in the *Zeitschrift* d. V. d. I., January 5, 1900. A general view of the *Felten* and *Guilleaume* electric starter for large engines is shown in Fig. 13–40, while Fig. 13–41 shows the diagram of connections and the method of operation for smaller sizes. The motor e by means of a chain drive operates the pulley s_2. Rigidly fastened to this pulley is the gear z_2. This gear in turn

* F. E. Junge, Power, April, 1906.

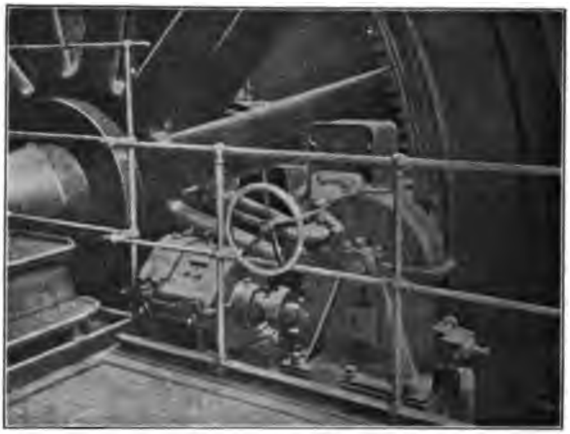


FIG. 13-40. — Felten & Guilleaume Electric Starter.

drives a gear z_3 which is pivoted on a swinging lever h_1 as shown. The arc z represents the pitch circle of the annular gear inside of the fly-wheel. Arrows indicate the direction of operation for each gear. One end of the lever h_1 is connected by means of a strong helical spring f to the toothed segment m. Meshed with this segment is the gear z_1 which is turned in one direction or the other by the lever h_2, which is the operating handle of the motor starting box a. In the diagram, lever h_2 has been pulled to the right as far as it will go. This action has started the motor e, and, by shoving the segment m to the left, has put the spring under considerable tension. This pulls the lower end of h_1 to the right and causes z_3 to mesh with the fly-wheel z. The gear z_3, being positively driven from z_2, starts the fly-wheel revolving. The entire mechanism is held in the position

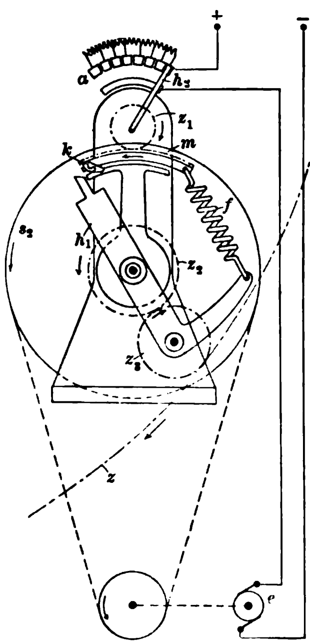


FIG. 13-41.

shown by a latch *k* at the end of the segment *m*. If now the engine picks up its cycle, and the wheel *z* attains a greater velocity than it can receive from z_3, the lever h_1 by the action of the fly-wheel will be pushed toward the left, putting the spring *f* under higher tension. This motion proceeds until the upper end of h_1 comes in contact with and unlatches *k*, when, under the influence of the spring, the segment *m* is pulled to the right very suddenly and the motor is shut off. At the same time the action of the fly-wheel upon z_3 throws this out of mesh with *z*. The entire labor connected with this starter therefore consists of the attendants turning the handle h_2 of the starting box to the right to start the motor. Beyond this no further attention is required, the action being entirely automatic.

CHAPTER XIV

REGULATION OF GAS ENGINES

In gas as in steam engines there are two kinds of speed variation. The first of these is directly due to the impulse of the explosion, and it manifests itself in a variation of the velocity of the crank pin during one engine cycle. To confine this speed variation to within the allowable limits set by the particular service to which the engine is to be put is the function of the fly-wheel. Fly-wheel computations are beyond the scope of this work, but it will be of interest to point out briefly the relation between the various types and combinations of engines as regards the fly-wheel weight required for any given service. This weight depends not only upon the closeness of regulation desired, but largely also upon the variation of the crank effort during one cycle or revolution, and this in turn depends upon the cycle employed and the cylinder combination.

The most complete exposition of this subject relating to gas engines is given by Güldner,* and from the data given by him the following table (see page 440) is adapted.

In deriving these figures it is assumed that the same coefficient of fly-wheel regulation, designated by δ_w, is maintained in all cases. This coefficient

$$\delta_w = \frac{V_{max} - V_{min}}{V},$$

where V is the mean velocity of the crank pin.

The second kind of speed variation above mentioned is the change in the number of revolutions of an engine in any given time due to a change of load. This variation is taken care of by the governor. Just as a limit is set to the speed variation during one cycle, so an allowable limit is set to the vari-

* Entwerfen und Berechnen der Verbrennungs-motoren, 2d ed.

TABLE OF RELATIVE FLY-WHEEL WEIGHTS FOR THE SAME COEFFICIENT OF REGULATION; FOR VARIOUS TYPES OF ENGINES AND CYLINDER COMBINATIONS. WEIGHT OF THE FLY-WHEEL OF A SINGLE-ACTING, SINGLE-CYLINDER, 4-CYCLE ENGINE IS TAKEN = 1.00.

Type of Engine and Cylinder Combination	Crank Travel Between Explosions, Degrees		Relative Fly-Wheel Weight for			
			Equal Cyl. Diameter, Stroke, and Revolution		Equal Maximum Horse-power	
	4-Cycle	2-Cycle	4-Cycle	2-Cycle	4-Cycle	2-Cycle
I. Single-Acting, Single Cylinder......	720	360	1.000	.802	1.000	.401
II. Double-Acting, Single Cylinder.....	540 & 180	180	1.230	.424	.615	.106
III. 2-Cylinder Tandem Single-Acting, 1 Crank...........	360	360	.796	1.595	.398	.399
IV. 2-Cylinder opposed, Single-Acting, 1 Crank...........	540 & 180	180	1.290	.335	.645	.084
V. 2 Cylinders in Parallel, Single-Acting 2 Cranks together	360	360	.792	1.602	.396	.401
VI. 2 Cylinders in Parallel, Single-Acting, 2 Cranks, 180° apart...........	540 & 180	180	1.290	.335	.645	.0.84
VII. 3 Cylinders in Parallel, Single-Acting, 3 Cranks, 120° apart...........	240	120	.678	.237	.226	.0395
VIII. 4 Cylinders in Parallel, Single-Acting, 4 Cranks, Nos. 1 & 3 together, Nos. 2 & 4 together, and 180° from Nos. 1 & 3......	180	—	.335	—	.084	——

ation that may occur in the number of revolutions from full load to no load, and the governor must be designed and set accordingly The allowable variation, the so-called coefficient of governor regulation, may be expressed by $\delta_r = \frac{n_{max} - n_{min}}{n}$, where n is the mean number of revolutions $\left(= \frac{n_{max} + n_{min}}{2}\right)$. What

the values of δ_w or δ_r may be depends altogether upon the kind of service, and the value of δ_r at least is, or should be, clearly stated in all engine specifications. The value of δ_w, that is, the variation in the angular velocity of the engine, can be made anything, depending upon the weight of the wheel put in. It usually varies from $\delta_w = \frac{1}{40}$, for ordinary commerical service, to $\delta_w = \frac{1}{225}$ say, for the most exacting service required for the operation of alternating-current generators in parallel. The value of the coefficient of governor regulation δ_r usually varies between $\frac{1}{50}$ and $\frac{1}{25}$, *i.e.*, the speed variation is from 2 to 4 per cent.

The theory of centrifugal governor design as applied to gas engines governing by throttling or cut-off cannot be taken up here, except merely to state that it is best, especially in the case of the power gases, to have the governor act upon the mechanism that positively operates the admission gear rather than upon the admission gear direct. In this way the governor does not in general have to be so powerful and above all it is unaffected by any clogging of the admission gear or valves due to tar and other incrustations. It should further be noted that any governor of this type that is too sensitive may react with the speed variation within one revolution and will thus be hunting constantly. Dash pots only forcibly overcome these conditions, and it is better to design these governors with plenty of adjustment regarding their sensitiveness.

The design of hit-and-miss governors (see below) is in general very simple. In contradistinction to the type of governors above mentioned, they can hardly be made too sensitive or astatic for the quicker they act, the better.

In the following we take up, first, Systems of Governing, and, second, Mechanical Details of Governors.

1. **Systems of Governing.** — At the outset an essential difference between steam and gas engines in the matter of governing should be noted. The working fluid in the steam engine is a comparatively stable medium, and, as long as the pressure remains constant, one position of the governor mechanism always corresponds to the same load, cycle after cycle recurring with the same development of power. This is absolutely essential for close governing, and in this respect the steam engine has some advantage over the gas engine. The conditions in

the latter are very different. The working fluid is prepared by the engine itself, air and fuel being mixed at the engine to produce the medium. Various expedients to this end, more or less successful, are in use, but outside of this, due to accidents of design or other reasons, stratification of the charge more or less complete, and variation in ignition, may result in unequal velocity of pressure propagation through the mass of the charge giving a bundle of different diagrams for the same heat value of the charge. Thus it may result that the same position of the governing mechanism may not, and often does not, indicate the same power developed, and speed fluctuations are the inevitable result. Fortunately the mixing and ignition apparatus of our modern engines can be made perfect enough in their action to confine these fluctutations to within allowable limits.

All gas engine governors come under the following systems as far as their effect upon the diagram is concerned. Mechanically they may be of various designs, as inertia, fly-ball, etc., as shown later:

I. The Hit-and-Miss System.

II. Variation of the Ratio of Fuel to Air with Change in Load; Quality Governing.

III. Variation of the Quantity of the Charge to suit the Load. Ratio Fuel to Air remaining constant; Quantity Governing.

IV. Combination Systems.

V. Governing by Varying Time of Ignition.

I. *The Hit-and-Miss System*

This system effects speed regulation by cutting out explosions altogether, depending on the load. Thus, for instance, if the engine is running at full load, the explosions or cycles will follow each other in regular order until the speed has increased enough above the mean to cause the governor to act, preventing the drawing in of the next charge, thus causing a "miss." This in turn causes the speed to fall sufficiently below the mean to make the governor act the opposite way, causing the explosions to recur. At any other load less than the full load the governor action is the same, except that as we go down in the scale the proportion of "misses" to "hits" constantly increases. This system may be operated in any of the following ways:

(*a*) By keeping the fuel valve closed, so that the engine draws only air for the miss cycle.

(*b*) By keeping the inlet valve closed, thus preventing the admission of both fuel and air.

(*c*) By keeping the exhaust valve open. In this case the admission valve is usually automatic, and its opening is prevented by the fact that on the next stroke no vacuum is formed, the exhaust gases being sucked back into the cylinder.

Theoretically this system of regulation is the simplest, and, from the standpoint of fuel consumption, the most economical; practically, however, it is beset with certain difficulties. In theory the cycles are all gone through under exactly the same conditions, and hence ratio of fuel to air, pressure of compression and point of ignition can all be adjusted once for all to suit the requirements of best thermal efficiency. The thermal efficiency of the cylinder should therefore be the same at all loads.

In practice there is some deviation from this ideal condition, even assuming *perfect governor action,* but the variation depends somewhat upon the manner of governing. Thus in engines in which only the fuel valve is kept closed to produce the miss cycles, it will generally be found that the card directly following a miss period is larger than those following it, at least for loads approaching full load. This is due to the fact that during the miss period the cylinder has been thoroughly scavenged by air, causing the next charge to be purer and somewhat larger in quantity than the average. Under very low loads the effect is apt to be the opposite, that is, owing to a prolonged period of miss strokes the cylinder has cooled so far as to make the first cycles following somewhat slow burning until the cylinder heats up again.

It is evident that these variations must have their effect upon cylinder efficiency, but the effect is perhaps greater with liquid fuel engines than with gas engines proper, because a cool cylinder is likely to condense some of the fuel vapor, thus causing a direct loss.

In engines that govern by keeping the exhaust valve open, drawing the exhaust gases back into the cylinder, the effects above outlined may be less marked, but the method cannot on that account be recommended as better than the other, because the inevitable mixing of the exhaust gases with the incoming charge has its own harmful effects.

In spite of these facts, however, the hit-and-miss system of governing, no matter how carried out, usually shows a somewhat greater economy of fuel in practice than the other systems.

We next turn to the efficiency of this system as a speed regulator. It is evident that the closeness of regulation in case centrifugal governors are employed depends altogether upon the sensitiveness of the governor, that is, upon the facility with which it changes from one position to the other; although it is possible here also to have a governor too sensitive, resulting in needless hunting. But whatever the type of hit-and-miss governor, the regulation will be closest if at the higher loads a constant series of explosions is followed by a single miss cycle, or if at the lower loads a single explosion is followed by a constant series of misses. Thus $\frac{3}{4}$ load should be represented by the series 111–111, etc., and $\frac{1}{5}$ load by 1 – – – – 1 – – – –, etc. Any disturbance of the governor, accidental or otherwise, as through want of care, increased friction, wear, etc., will alter this ideal condition so that a $\frac{3}{4}$ load, for instance, may be represented by the series 111 – 11 – 111 – 11, etc. But such variation at once unfavorably affects the regulation.* These accidental conditions are not under the control of the designer, and not always under the control of the operator, and the net result is that hit-and-miss regulation, though economical, is somewhat unreliable and certainly not as close as that obtained by some of the other methods, unless a very heavy fly-wheel is employed.

Hit-and-miss governing is therefore little employed where close regulation is essential, as for electric current generation. For ordinary commercial power operation, where the regulation need not be closer than say 3 to 5 per cent., the system is quite satisfactory, although it is being slowly replaced even in this field. It should be remembered in this connection that, if the engine is belt-connected to the power consumer, the flexible connection will tend to equalize the speed variations to a certain extent.

II. *Governing by Varying the Ratio of Fuel to Air: Quality Governing*

In this system the governor is usually made to act upon the fuel admission valve, so that as the load on the engine decreases

* See Mollier, Zeitschrift d. V. d. Ingenieure, 1903, p. 1704.

the engine receives less and less fuel in the same total charge volume. This of course decreases the area of the indicator card developed to suit the load. Instead of acting upon the fuel valve, this method of governing has also been carried out by sucking back a certain amount of the exhaust gases, thus also decreasing the heat content of the charge. Another way is to regulate the air admission valve, making the fuel valve automatic. All things considered, however, the first mentioned method is the best.

Considered from a thermal standpoint this system has the advantage that, since the total charge volume remains practically the same for all loads, the compression pressure remains constant throughout. The cards obtained will therefore be somewhat as

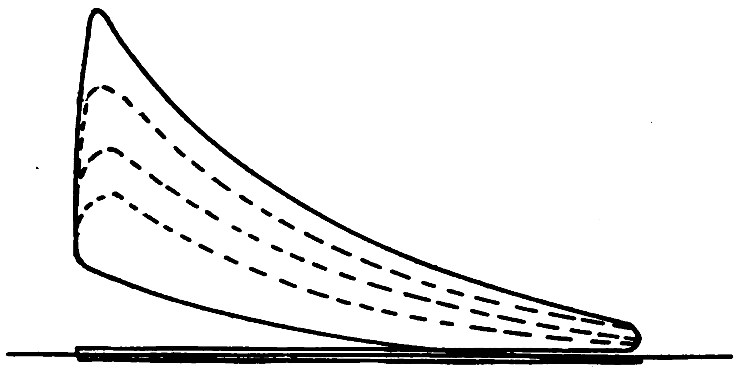


Fig. 14–1.

shown in Fig. 14–1, in which the full line represents the full load diagram. It therefore should follow on theoretical grounds that the thermal efficiency of the cylinder should be about the same for all loads. In practice, however, it has been clearly shown that this system is inferior at low loads to the next one to be described. In fact, the fuel consumption per horse-power usually increases very rapidly as the load drops. The reason is that, as the fuel-ratio is decreased, the mixture rapidly becomes difficult to ignite and, above all, slow burning. This necessarily increases the heat loss to the jackets and the ignition difficulty may go as far as to prevent ignition altogether, causing a direct loss of fuel. In most cases after-burning is clearly recognizable by the slow dropping of the expansion line. Designers have tried to over-

come this difficulty by placing the time of ignition also under control, making it earlier as the load decreases. The scheme, however, does not appear to have been very successful.

As a method of governing, this system is capable of giving close regulation with the proper weight of fly-wheel. The very fact, however, that the compression pressure does not drop in proportion to the maximum pressure introduces a disturbing factor into the crank effort diagram which would tend to make the regulation under this system less close at low loads than under System III.

III. *Governing by Varying the Quantity of Charge of Constant Composition to suit the Load: Quantity Governing*

Governing by changing the quantity of charge to suit the load may be carried out in three ways:

(*a*) The engine draws a charge full stroke each time, but a part of the charge, depending upon the load, is forced back into the suction passages, the inlet valve being under governor control.

(*b*) The incoming charge is completely cut off by the governor at the proper time, the charge expanding behind the piston for the rest of the stroke. This is known as the cut-off method.

(*c*) The charge is throttled down throughout the entire suction stroke, the governor determining the position of the inlet valves. This is called the throttling method.

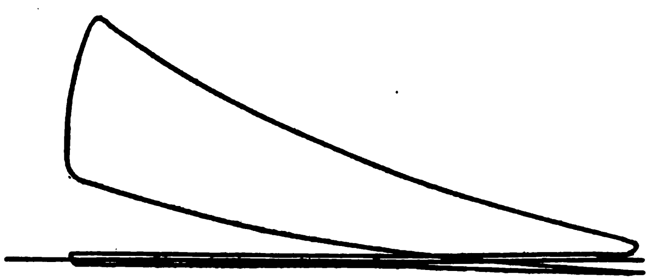


Fig. 14–2.

Figs. 14–2 and 14–3 show the differences in the diagrams obtained under conditions (*b*) and (*c*) outlined above.

Quantity governing in general is, on thermal grounds, open to the objection that the compression pressure decreases with the load, and hence the cylinder efficiency constantly decreases. On the other hand, the mixtures remain readily ignitable down to the friction load, with the result that quantity governing is on the whole more economical than quality governing. The fact, too, that the compression pressure decreases with the maximum pressure has a favorable influence upon the crank-effort diagram, admitting of close regulation.

As between methods (*b*) and (*c*), the former is slightly better because less work is lost in the lower loop. Everything considered, where close regulation is essential, the cut-off method of quantity governing is the best.

Fig. 14–3.

Regarding the economy of the cut-off method as compared with the hit-and-miss method of governing, E. Meyer* finds that down to about ½ load the two systems are about on a par. Below this load the efficiency of the cut-off as compared with the hit-and-miss method rapidly falls off.

IV. *Combination Systems*

It has been attempted to perfect quantity regulation by changing the compression space so as to keep the compression pressure the same at all loads. Thermally this is a step in the right direction, but no successful machine operating upon this system has yet appeared.

Another combination system is that of Letombe, which is in successful use. Letombe regulates by lengthening the time of

* Zeitschrift d. V. d. Ingenieure, April 25, 1903.

opening of the inlet valve but decreasing the lift of the gas valve as the load decreases. As far as the fuel is concerned, this is quantity regulation, but the longer time of opening of the inlet valve increases the total charge volume, which means that the leaner mixtures will be more highly compressed than those for higher loads. This is thermally correct. Another point is that the richer mixtures at the higher loads, although less highly compressed, are less in total volume than the leaner mixtures. Hence as the load increases the ratio of expansion increases as compared

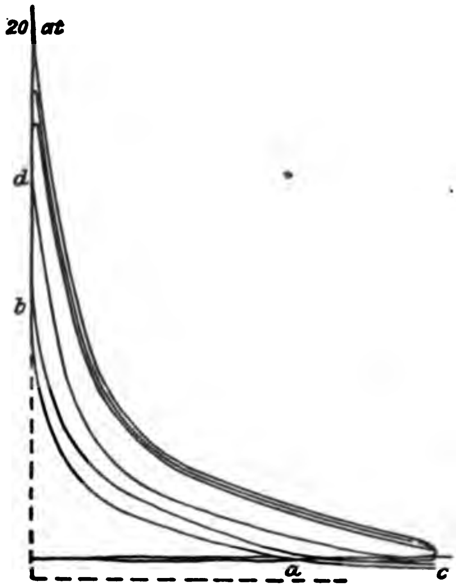


Fig 14–4.

to the ratio of compression, which tends to draw down the terminal pressure at the end of expansion and decreases the exhaust loss. The resulting diagrams are shown in Fig. 14–4, given by Güldner. The compression line *a–b* belongs to the full load card, line *c–d* to the minimum load card. The latter card shows suction full stroke and a compression pressure of about 190 pounds, the former shows a suction volume equivalent to about 55 per cent stroke and a compression pressure of about 115 pounds. In spite of the thermally excellent features, which gives good

regulation, the economy regarding fuel is no greater than that obtained by a purely cut-off system.

Other combination systems that have been employed are quantity regulation at high loads combined with quality regulation at low loads or vice versa. This is done in some German engines. To compensate for the slow burning of the leaner mixtures the spark is advanced. Lastly, engines that govern either by quantity or quality regulation at the higher loads have been governed by the hit-and-miss method at very low loads. This is done in the American-Crossley engines and also by Letombe in the system above described.

V. *Governing by Varying the Time of Ignition*

Strictly speaking, the time of ignition should be adjusted to suit the kind of charge. That means, for instance, that in quality regulation the spark should be advanced with a decrease of load. It has already been mentioned that this has been tried both by governor control or by hand regulation. The former is rather difficult because proper ignition is subject to so many accidental variations, but hand control is quite practicable and all stationary engines should therefore be furnished with adjustable spark gear.

Automobile engines are generally governed by hand regulation of the throttle in combination with the spark.

Governing of Two-Cycle Engines

Small two-cycle engines are usually governed by throttling either the fuel or the charge. In the first case this results in what is practically quality regulation, in the second in quantity regulation. Liquid fuel engines of this type nearly always govern by adjusting the stroke of the pump to suit the load, resulting in quality regulation. In the larger machines, which are nearly always served by separate pumps, it is absolutely essential that the cylinder be thoroughly scavenged. Hence it is usual to first admit air alone directly from the pump or an intermediate receiver, and a little later the fuel or the mixture as the case may be, the point of admission of the latter being under governor control. The governor may act either on the inlet valve or directly on the pump. It should be noted that if reservoirs

are used between pump and engine, there is likely to be a lag of several strokes between the action of the governor and its effect on the engine. For more detailed information on the governing of large two-cycle engines, see the descriptions at the end of this chapter.

2. **Mechanical Details of Governors.** — For hit-and-miss governing, inertia governors of the so-called pendulum type are extensively used. Besides this, centrifugal governors of the fly-ball type also find application. For the "precision" systems of regulation, *i.e.*, quantity and quality regulation, centrifugal governors of the fly-ball type are mostly used; shaft governors either of the centrifugal or inertia type are, however, lately finding application.

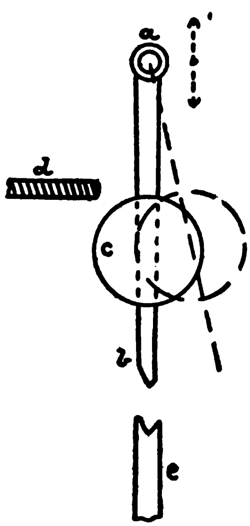


Fig. 14–5.

(*a*) Pendulum or Inertia Governors for Hit-and-Miss Regulation. — The simplest form of this governor is shown in Fig. 14–5. The rod *a–b* carries the ball weight *c*, and receives an up-and-down motion usually from the lay shaft as shown by the arrows. As the rod moves down, the weight strikes the projection *d*, and the pendulum is thrown to the right as shown by the dotted position. If the speed is right, the pendulum will have gone back to its normal position by the time the point *b* reaches the end of the valve stem *e*, and the gas or inlet valve will be opened. If the speed is above the normal, *a–b* will not be back in its normal position, and *b* will miss the valve stem *e*, causing a miss-stroke.

A modification of this, the bell-crank pendulum, is shown in Fig. 14–6. The action of this governor is clear from the previous description.

The disadvantage of these two forms may be said to be that they are thrown a considerable distance out of their normal position. This is overcome in what may be called spring-loaded pendulum governors, the fundamental type of which is shown in Fig. 14–7. As the bell crank *a b c* moves downward, the inertia

of the weight, *c*, throws the governor into the dotted position. If by the time the stem of the valve, *e*, is reached, the spring has failed to pull the governor back into its normal position, the valve will not be opened, causing a miss-stroke.

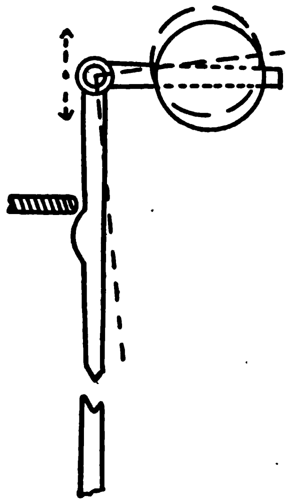


FIG. 14-6.

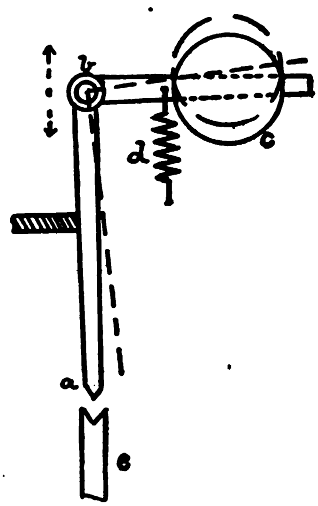


FIG. 14-7.

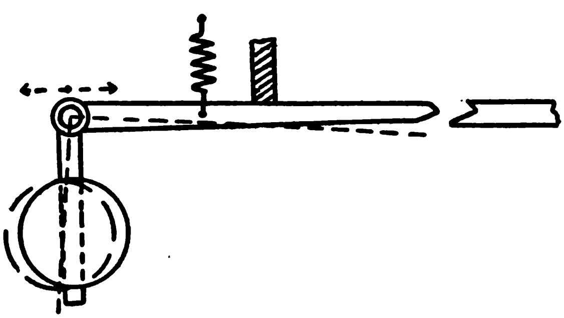


FIG. 14-8.

Fig. 14-8 shows the simplest kind of inertia governor for a horizontal valve stem. This may be used with or without the spring. Its action is sufficiently plain.

There are many modifications of these fundamental types.*

* See Güldner, Entwerfen & Berechnen der Verbrennungsmotoren, p. 354.

These will readily suggest themselves to suit any particular case. The following are some examples from actual practice.

Fig. 14–9 shows a modification of the vertical pick blade governor that has been used on Crossley engines.* The lever, *a*, is

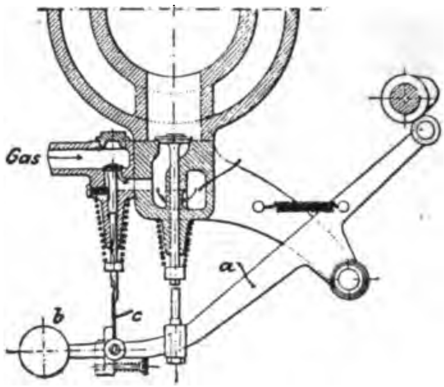


FIG. 14–9.

actuated by the cam, and as long as the speed is normal the spiral spring on one arm of the bell crank is strong enough to make the

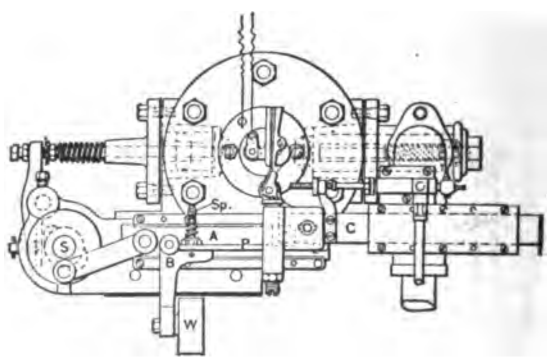


FIG. 14–10.

weight, *b*, attain its normal position in the time available. The blade, *c*, then opens the gas valve. If the speed rises above normal, the inertia of the weight, *b*, throws it down so far that it cannot reach its normal position in time and *c* misses the valve stem.

* Güldner, Entwerfen & Berechnen der Verbrennungsmotoren.

The governor on the Springfield engine, Fig. 14–10, is quite similar to the above, except that the blade acts horizontally.* In this case the bell crank is carried by the slide, *A*, which is actuated from *S*. For normal speed the blade, *P*, is horizontal, and in this position hits the stem of the gas valve. For excessive

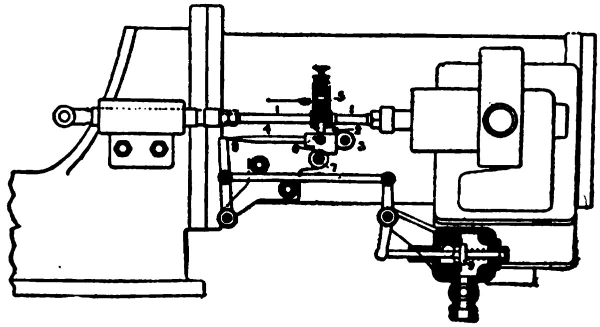


FIG. 14–11.

speed, the spiral spring fails to bring the weight *W* back into normal position in time and *P* misses the gas valve.

An English design of hit-and-miss governor not quite so simple is shown in Fig. 14–11.† Here the inlet valve stem, 1, carries the

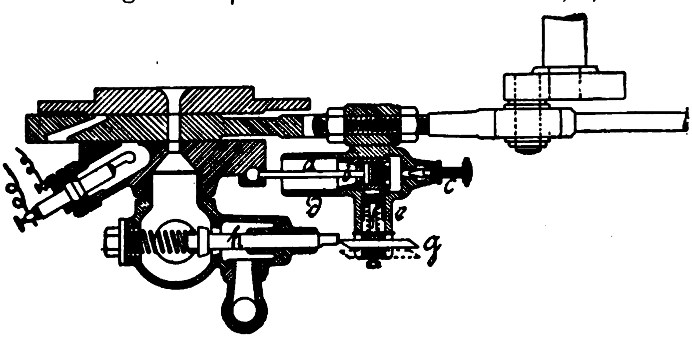


FIG. 14–12.

bracket, 2. To this is pivoted a blade, 4, at 3. For normal speed of the engine, the spring, 5, gently presses the inclined surface, 6, against the stationary roller, 7. In this position the blade, 4, will hit the lever, 8, thus opening the gas valve, 9. Should the

* Power Quarterly, Oct. 15, 1900.
† Clerk, Gas and Oil Engines.

speed, however, rise above normal, the upward throw given to the blade due to the inclined surface, 6, sliding against the roller, 7, becomes excessive, and blade 4 fails to assume its normal position by the time the lever 8 is reached. The gas valve then fails to open.

An ingenious modification of the above types of hit-and-miss governors used by Delamare* is shown in Fig. 14–12. The stem of the inlet slide valve carries a small cylinder, *a*, into which is fitted a stationary piston, *b*, air-tight. The needle valve, *c*, serves to adjust the velocity of air escape from behind the piston, *b*. With the cylinder, *a*, in the extreme right-hand position, air is admitted behind *b* through the groove, *d*. As *a* travels to the left at the proper speed, the air is compressed, but it escapes through the needle valve, *c*, just fast enough to prevent its pushing outward the plunger, *f*, in the branch cylinder, *e*, against the spring. In this position the blade, *g*, hits the gas valve stem, *h*. For an excess in engine speed, cylinder *a* travels to the left so fast that the air cannot escape at the proper rate, the pressure generated forces the plunger, *f*, outward, and *g* misses the valve stem as shown in the dotted position.

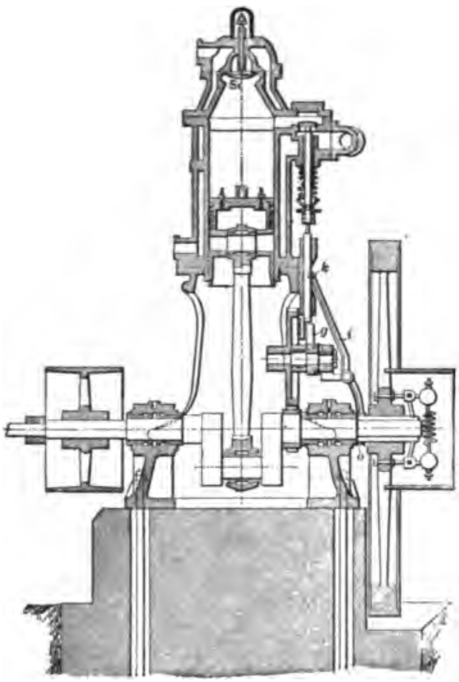


FIG. 14–13.

(*b*) CENTRIFUGAL GOVERNORS FOR HIT-AND-MISS REGULATION. — A hit-and-miss governor of this type is used in the French engine shown in Fig. 14–13.† The centrifugal governor

* Schöttler, Die Gasmaschine.
† Power Quarterly, Oct. 15, 1900.

in the fly-wheel, when the speed becomes too high, actuates the latching arrangement *i–k* and throws the oil pump serving to charge the vaporizer temporarily out of action.

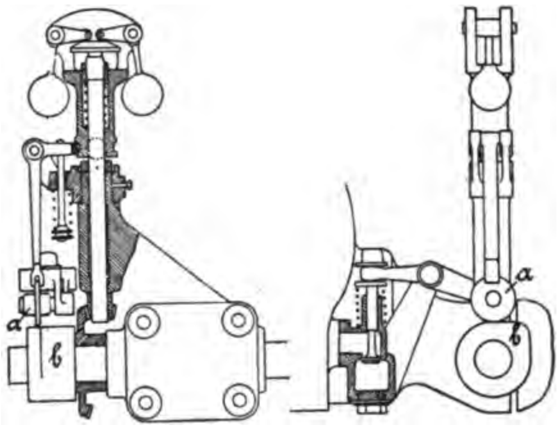


FIG. 14–14.

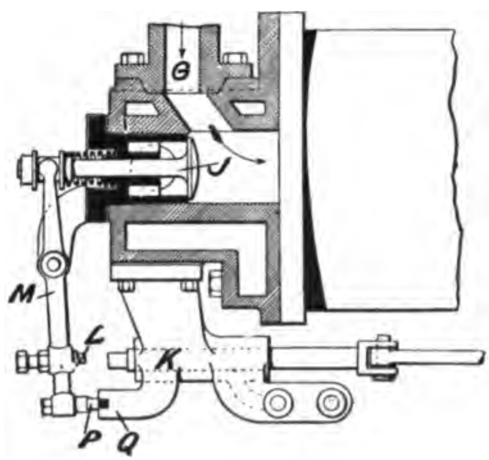


FIG. 14–15.

The Robey governor,* Fig. 14–14, is of the fly-ball type; normally the roller, *a*, under the control of the governor travels on the cam, *b*, thus opening the gas valve at the proper time. Under excessive

* Schöttler, Die Gasmaschine.

speed, however, the roller is shoved to the left on its spindle by the governor linkage, and misses the cam.

The Campbell oil engine has a fly-ball governor which operates to keep the exhaust valve open when the speed exceeds the normal, Figs. 14–15 and 14–16.* The rising of the governor weights, Fig. 14–16, depresses the end, *O*, of the lever, *N*, thus interposing the plate, *P*, Fig. 14–15, between the end of the exhaust lever, *M*, and the stationary bracket, *Q*. This prevents the exhaust valve from closing until the dropping of the speed causes the governor

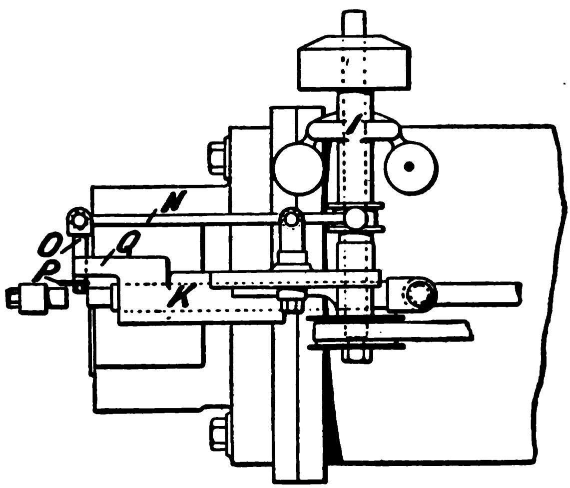


Fig. 14–16.

to withdraw the plate, *P*, when the exhaust valve is again regularly opened by the sliding piece, *K*.

A shaft governor acting to hold the exhaust valve open is used on the Perkins engine, Fig. 14–17.† The exhaust valve is operated by the lever *R*, which in turn is actuated by the block *A*, striking the end block, *B*, of the rod, *C*. As the speed exceeds the normal, the governor weights move toward the circumference of the wheel, and, through suitable linkage, throw the latch, *P*, into a recess in the side of the block, *B*. This prevents the rod, *C*, from returning, and keeps the exhaust valve open.

* Clerk, the Oil and Gas Engine.
† Power Quarterly, Oct. 15, 1900.

(*c*) Quality Regulation.—The quality method of regulation, as carried out for gas fuel, is best exemplified in the Nürnberg engine. This engine has separate gas and inlet valves. At all

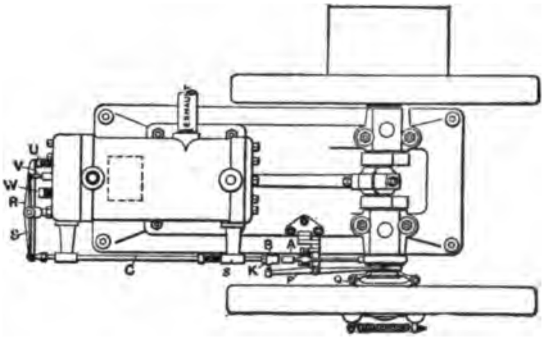


Fig. 14–17.

loads the inlet valve opens and closes at the same time, thus admitting a constant charge of volume. But the gas valve, under governor control, opens later as the load drops, and closes about the same time as the main inlet valve. Thus the composition or

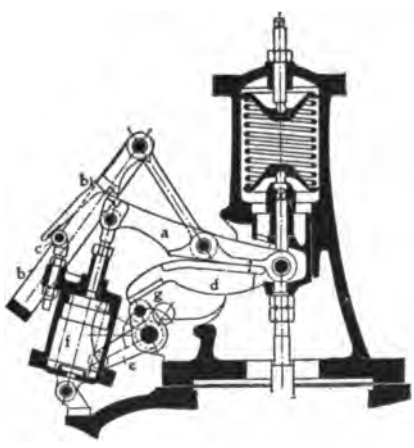


Fig. 14–18.

quality of the mixture changes with every load. A certain disadvantage of this method of proportioning the charge is that the mixture is not only affected by the relative time of opening of

the gas and inlet valves, but also by the relative velocities of the gas and air currents. Thus at low loads, when the gas valve opens, the gas column has to start from rest, while the air column already has its maximum velocity. The time needed for the acceleration of the gas column is practically constant, hence this factor affects the mixture differently for every load. Another point is that unless the gas valve closes quickly there is apt to be so much throttling of gas as to make the mixture drawn in at the end for low loads incombustible. This is especially bad since this mixture is apt to be located around the igniter.

The governor used on the Nürnberg engine is of the ordinary fly-ball type. The trip gear for the gas valve is shown in Fig. 14–18.* The governor determines the position of the bell-crank

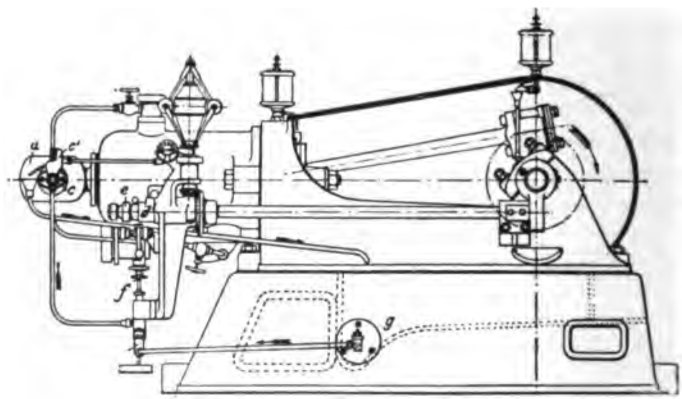


FIG. 14–19.

lever *e–g*, and through this the position of the wiper cam lever ***d***. The eccentric rod, *b*, through the latch b_1, lifts the gas valve by depressing the valve lever, *a*. The position of ***d*** determines the time of opening. The valve is closed practically instantaneously by the coil spring shown in the top of the housing as soon as the small roller, *c*, has released the latch b_1. The dash pot, *f*, serves to dampen the drop of the valve, making it close without shock.

Many oil engines are governed by what is practically quality regulation. Thus the Hornsby-Akroyd oil engine, Fig. 14–19,† takes a full charge of air every stroke, but the quantity of oil is

* Power, Feb., 1906.

† Güldner, Verbrennungsmotoren, p. 119.

changed to suit the load. The quality of the mixture therefore changes for every different load. The proportioning of the oil supply to the load is accomplished as follows. The oil pump supplies a constant quantity of oil every stroke to the vaporizer valve, *c*. This valve, however, is a two-way valve, one exit opening through an atomizing nozzle into the vaporizing chamber, while the other allows some of the oil to flow back into the oil tank. The fly-ball governor, through the linkage shown, controls the size of this overflow opening, *c'*, and thus determines the amount of oil which enters the vaporizer chamber.

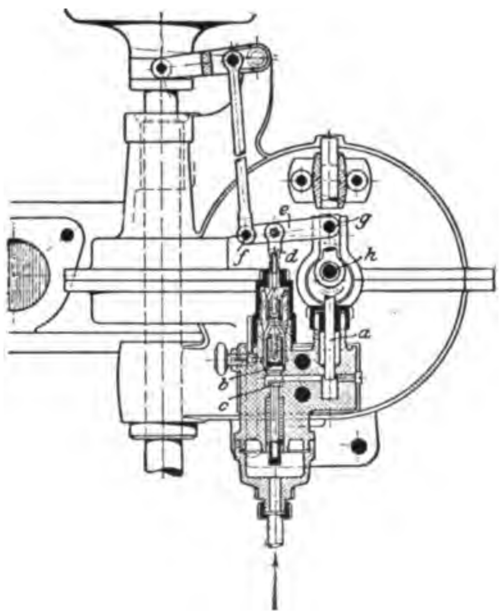


FIG. 14-20.

The governing mechanism of a small Diesel oil engine is shown in Fig. 14-20.* The pump plunger, *a*, is actuated by the lay shaft, *h*. The suction valve is shown at *c* and the automatic discharge valve at *b*. The oil under pressure flows through *b* to the atomizing nozzle. The amount of oil required for the load on the engine at any given time is measured by controlling the time of closing of the suction valve, *c*, forcing more or less of the oil back into the

* Güldner, Verbrennungsmotoren, p. 379.

oil supply tank. The motion of valve *c* is controlled through linkage not shown, by the governor lever *e*. At *g* this lever receives an up-and-down motion from the lay shaft, but the manner in which this motion is transmitted to the suction valve depends upon the position of the fulcrum, *f*, which is determined by the governor, thus varying the time of closing of the valve.

(*d*) QUANTITY REGULATION.—Koerting Bros. govern their four-cycle engine as follows, Fig. 14–21*: Air enters through the pipe, *D*, and gas through *B*. The two are mixed in the proportion suited to the particular gas by the mixing valve *A*. This valve

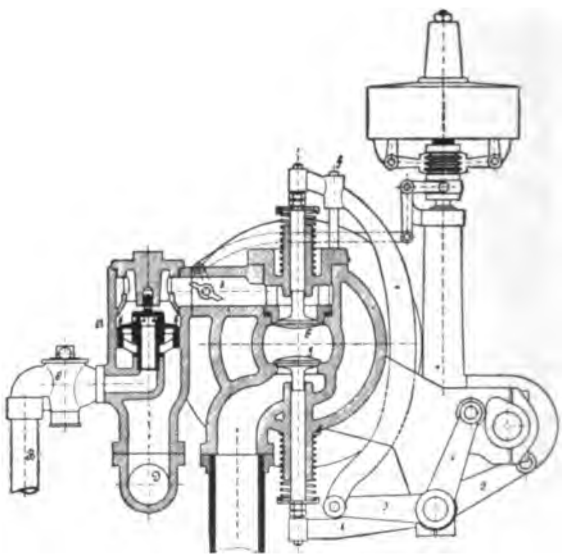


FIG. 14–21.

when once adjusted therefore furnishes a mixture of constant quality. To suit the quantity of this mixture to the load, the governor controls the position of the throttle valve *e* in the passage leading from the mixing valve to the inlet valve. This valve thus throttles the charge during the entire suction stroke.

A very similar arrangement is used in the Westinghouse vertical engine, Fig. 14–22.† *A* is the throttle valve controlling the entrance of the mixture to the cylinder. Its position depends upon the

* Schöttler, Die Gasmaschine.

† L. S. Marks, Instruction Paper on Gas and Oil Engines.

action of the fly-ball governor, *B*. Gas enters the interior of this valve through ports *G* and air through ports *D*. The relative proportions of these ports are fixed by slides *H–H*, so that the proper relation of air to gas for the fuel used can be established and maintained.

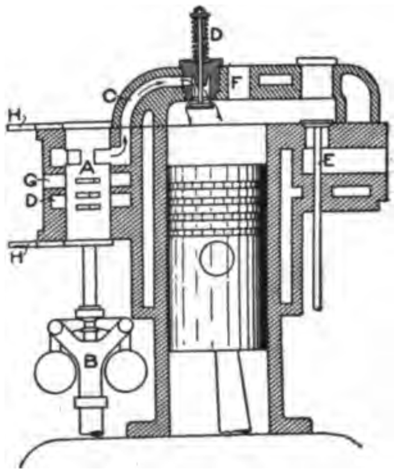


Fig. 14–22.

Fig. 14–23 shows the arrangements for operating and controlling the mixing valve in the Ehrhardt and Sehmer engine, which is a modification of the Deutz engine.* It will be readily seen, by a study of the figure, that the governor, through its linkage, controls the position of the fulcrum, about which the valve lever turns. The farther this fulcrum moves to the right, the greater will be the opening of the inlet-valve and the greater, consequently, also, the amount of mixture, proportioned in passing the valve, admitted to the cylinder.

It will be noted that all of the above examples of quantity governing throttle the mixture throughout the suction stroke. It was pointed out, in the discussion on various systems of governing, that the system employing quantity regulation with cut-off was superior to that which throttles the charge throughout the stroke, and it seems surprising, therefore, that this system is not in more extended use. There is to the writer's knowledge only

* K. Reinhardt, Bi-monthly Bulletin of the Am. Inst. of Mining Engineers, November, 1906, p. 1092

one engine in this country employing the cut-off system, the Sargent complete expansion engine. This engine employs a Rites inertia governor which acts upon the lay shaft driving gear which is loose on the crank shaft. As the load increases or decreases, the governor retards or advances this gear relative to the crank shaft, thus affecting all of the valve events depending upon the lay shaft at the same time. Since near the end of the stroke, however, the piston velocity is comparatively small, a relatively

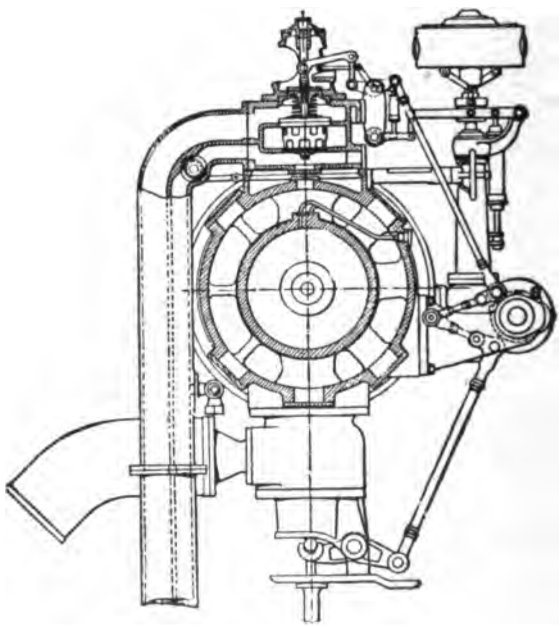


FIG. 14-23.

large advance or retardation of the gear will not affect the exhaust events or the beginning of the suction stroke very much. But the cut-off occurs along the suction line somewhere near the time when the piston has its maximum velocity, and hence a relatively small advance or retardation of the gear is sufficient to change the cut-off materially and hence to control the speed.

(*e*) COMBINATION QUANTITY AND QUALITY REGULATION. — The purpose of these combination systems has already been

pointed out. Fig. 14-24 shows the method as carried out by Reichenbach.* The governor, through the rod, *c*, controls the position of the bell crank lever, *d*. This lever has two slotted arms, *a* and *b*. The former, by rod and lever, regulates the position of the mixing valve, while the latter, by a similar arrangement, controls the position of the throttle valve. Suppose now that the sliding blocks *a* and *b* are in the position shown in the figure. Under these circumstances, the movement of the throttle valve is insignificant

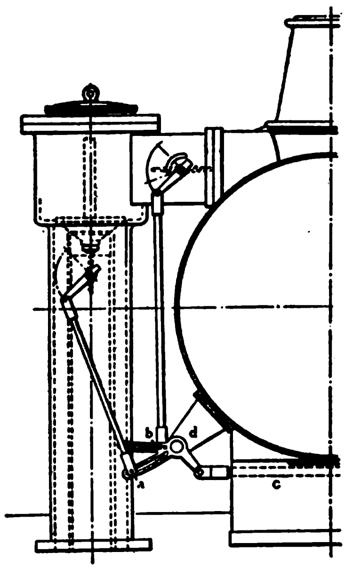


FIG. 14-24.

because *b* is so close to the center. Hence in this position the engine is practically governed by quality regulation. If the positions of *a* and *b* were reversed, that is, *b* at the outer end of its slot and *a* at the inner end, there would be practically quantity regulation. In actual operation the positions of *a* and *b* in their respective slots are determined by trial and depend upon the gas used. The final adjustment is such that for the upper ranges of load the quality of the mixture is changed, while for the lower loads the throttle valve comes into action while the mixture is

* Power, July, 1906.

held at practically constant composition. In addition to this. Reichenbach also puts the point of ignition under governor control, advancing it as the load drops.

Reinhardt's method of governing does not come strictly under any of the heads so far discussed. The inlet valve is shown in Fig. 14-25.* The valve opens and closes with the beginning and end of the suction stroke. Above the valve there are arranged a series of ports, I for gas and II and III for air. The flow of gas

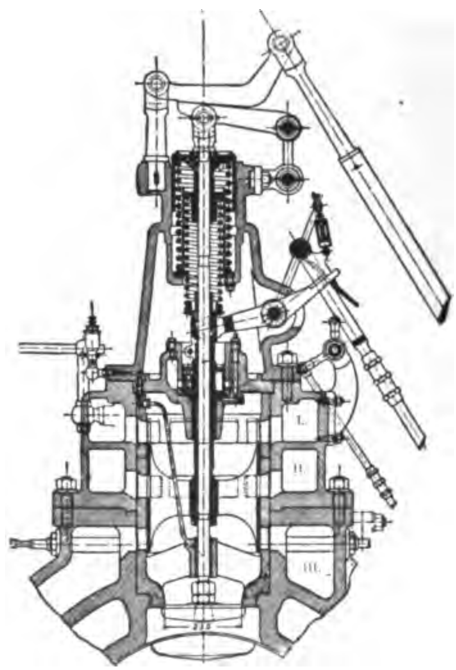


FIG. 14-25.

or air from these ports into the valve chamber proper is controlled by a slide guided by the valve spindle. At the beginning of the suction stroke the slide only frees the ports from chamber III, and hence only air is admitted. At some point in the stroke, however, as determined by the governor, the slide is suddenly released and forced down by springs. This closes ports III and opens gas ports I and air ports II. Mixture of the proper proportion then

* Reinhardt, Bi-monthly Bulletin, Am. Inst. Mining Engs., Nov., 1906.

enters the cylinder until the end of the suction stroke. As far as the admission of a varying quantity of constant mixture is concerned, this system is pure quantity regulation; but since the amount of air drawn from III increases as the load decreases, the mixture in the cylinder as a whole grows leaner as the load drops. It is claimed, however, that combustion is good even under friction load, because although the amount of constant mixture drawn in is small, more or less stratification of the charge ensures the presence of a fairly rich mixture around the igniter. The system has the advantage over pure quantity regulation in that the compression pressure remains practically unchanged.

In general principle the above method of Reinhardt seems to be a modification of the method of Mees patented in 1901.

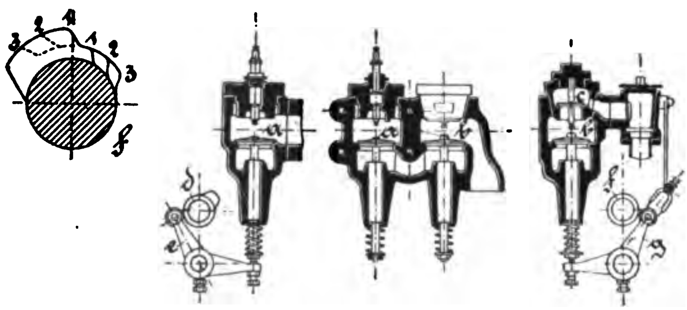


FIG. 14–26.

Letombe, recognizing the serious effect on thermal efficiency of the decrease in compression which accompanies pure quantity regulation, goes one step further and so arranges the valves that the lean mixtures under low loads are more strongly compressed than the rich mixtures at the higher loads. This is thermally an important step in the right direction. In Fig. 14–26,* *a* is the main inlet valve operated at a constant lift by the cam, *d*, through the lever, *e*. Ahead of this inlet valve are the air valve, *b*, and the gas valve, *c*. Valve *b* is operated through the lever, *g*, by the cam, *f*. Gas valve *c* opens only when pushed up by *b*. Cam *f* is shown in greater detail at the left. It has a number of elevations, each of which consists of two steps of different lengths, as shown at 1, 2, and 3. At full load the roller on the lever, *g*, mounts

* Schöttler, Die Gasmaschine.

the higher step of elevation, 1, opening both *b* and *c* wide, and the mixture, of constant proportion, enters the cylinder through *a*, which has been opened at the same time. After awhile the roller drops to the lower step of elevation, 1, air valve, *b*, closes partly, closing gas valve, *c*, completely. Only air is then drawn into the cylinder until the roller, *g*, drops off the cam altogether. Now suppose that the load drops. The governor then pulls over the roller, *g*, until it comes in line with elevation 2 or 3, as the case

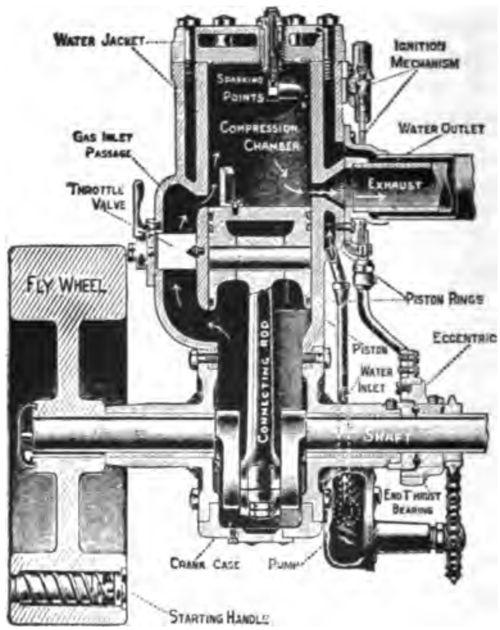


Fig. 14–27.

may be. The time of opening of the gas valve, *c*, is then shortened, because the higher step of the elevation is shorter; but the time of admission of air alone to the cylinder is lengthened, because the lower step of the cam is longer than it was before. Hence, although the mixture is leaner, the total charge volume is greater than at full load, and the leaner mixtures therefore receive a higher compression. Theoretically, this should give increased thermal efficiencies at low loads as compared with other systems

of regulation. In practice, however, for reasons not explained, there does not seem to be much difference.

(*f*) GOVERNING OF TWO-CYCLE ENGINES. — Fig. 14–27 shows a type of two-cycle engine, the Lozier,* much used for small motor boat work. As in all machines of this kind, the mixture is formed and compressed in the crank-case, and flows from here through a communicating passage into the cylinder, as shown by the arrows. Speed is controlled simply by the throttle valve in the passage, thus controlling the amount of mixture entering the cylinder. This is a very simple case, because the load on the engine is practically constant.

In the new Buckeye two-cycle engine, which, as far as size is concerned, is intermediate between the small two-cycle gasoline engine and the very large two-cycle blast furnace gas engines of the Koerting or Oechelhäuser type, the forward end of the cylinder is used as a mixture pump. Regulation is effected by means of the balanced throttle valve 60, Fig. 12–31, p. 289, which, under the control of the fly-ball governor, regulates both the suction to the pump and the delivery from the pump to the combustion chamber. Thus this engine governs by varying the quantity of the mixture; but since the cylinder is always thoroughly scavenged by fresh air, the mixture will naturally grow leaner as the load drops, while the compression pressure remains practically the same.

The governing details of the Oechelhäuser two-cycle machine vary somewhat, depending upon the firm building them. Fig. 14–28 † shows the scheme adopted by Borsig. It may be remembered that in this machine air and mixture enter the cylinder through separate rings of ports in the side of the cylinder, as soon as the piston uncovers them. The air ports are uncovered first to admit the scavenging air, the mixture ports a little later. Each ring of ports is surrounded by a receiver into which the air and gas pumps deliver their charges under some compression. It is evident that, since the time taken by the piston to uncover and cover the gas and air ports is practically constant, the amount of mixture that can enter the cylinder during a given time must depend upon the pressure in these receivers, and upon the port area.

* Power Quarterly, Oct. 15, 1900.

† Riedler, Gross-Gasmaschinen, p. 67.

The governor, therefore, is made to act upon the gas and air pumps to control the pressure in the receivers. Further, as the load decreases and approaches the friction load, the amount of mixture entering becomes so small as compared with the amount of air in the cylinder that ignition is likely to become difficult. For this reason the mixture ports are surrounded by a slide under governor control, as shown at *A*, Fig. 14–28, which is operated in such a way that as the load decreases the ports opposite the igniter are

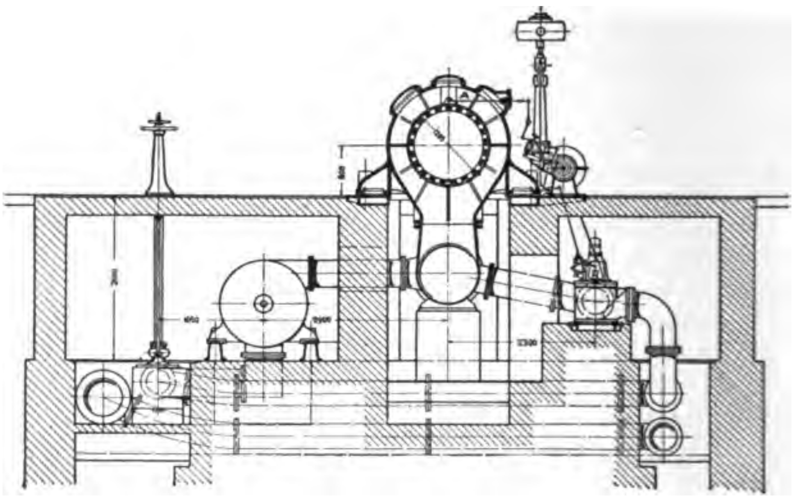


FIG. 14–28.

gradually closed first, thus insuring a comparatively rich mixture around the igniter at all times.

The method of governing the Koerting two-cycle engine also differs somewhat, depending upon the manufacturing firm. In all designs there are a gas and an air pump, of which the air pump delivers its charge into the cylinder from the commencement of its discharging stroke in order to scavenge out the cylinder. The gas pump runs idle for a part of its discharge stroke, generally forcing the gas back into the suction main, until, at a point determined by the governor, the overflow valve closes, and the gas pump delivers the rest of its charge into the cylinder through a valve on top. Fig. 14–29 * shows the pump construction used by

* F. E. Junge in Power, November, 1906.

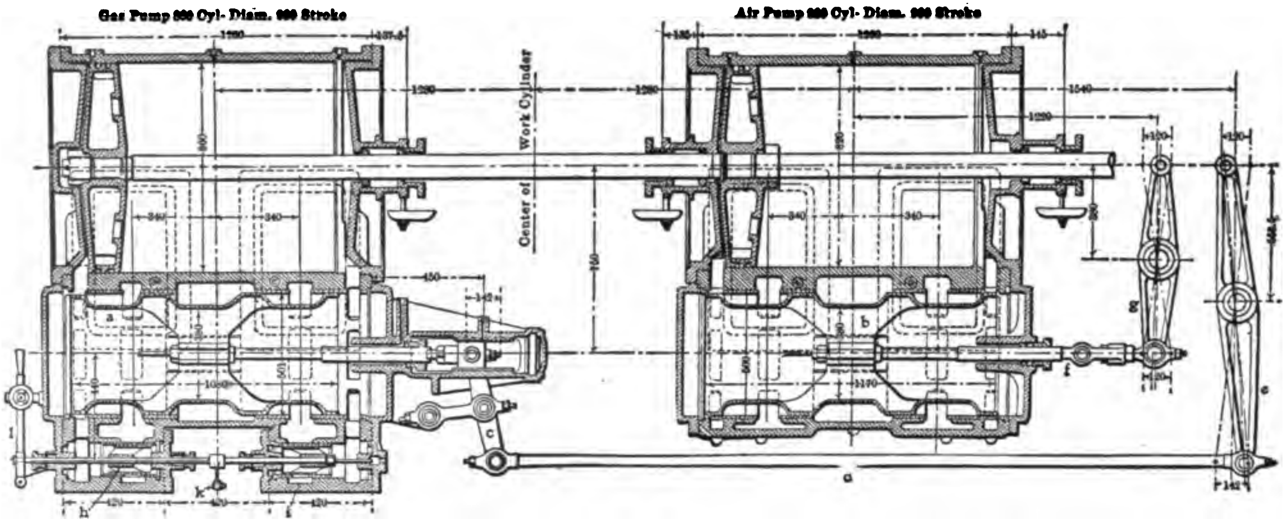


FIG. 14–29.

the Gutehoffungshütte. Both the gas and air pumps are controlled by piston valves operated by rocker arms and eccentrics. Situated below the piston valve of the gas pump there are two

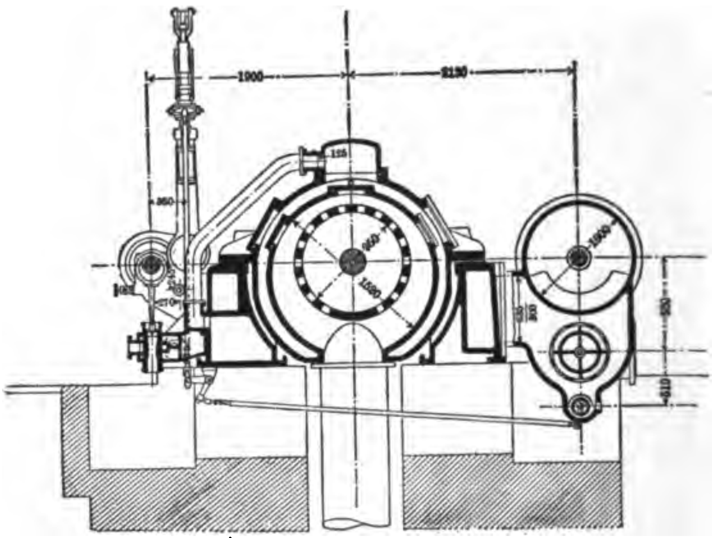


FIG. 14–30.

auxiliary piston valves, *h* and *i*, which are under the control of the governor. Depending on the load on the engine, the governor, through linkage shown in Fig. 14–30, and of which lever *k*, Fig. 14–29, is a part, rotates these auxiliary valves about their axis, thus controlling the time when the gas pump commences to deliver its charge to the power cylinder.

CHAPTER XV

THE ESTIMATION OF POWER OF GAS ENGINES

It is intended in this chapter to point out the various methods in use by which the power that a given engine may be expected to develop can be computed, or, what amounts to the same thing, to determine the cylinder dimensions for any given power.

In steam engine practice this is a comparatively simple matter. It is necessary merely to lay down an ideal indicator card along well-defined lines, and then, by the use of card factors closely fixed by long practical experience, to determine the probable mean effective pressure in the cylinder or cylinders.

Although this method is by some writers advocated also for the gas engine, the determination of the card factor for gas engines is based upon so many component factors, which in turn depend altogether upon the judgment of the designer, that the result is anything but certain. The difficulties attaching to this method are pointed out in greater detail below.

Before describing the various methods of computing cylinder dimensions it is well to examine briefly into the allowable piston speed. The theoretical limit of piston speed of course depends directly upon the time of explosion, and this in turn depends upon the kind of mixture and to a certain extent upon the method of ignition. As far as modern practice goes the upper limit to-day seems to be at about 800 feet per minute. A certain formula, empirical as far as the writer is aware, makes the piston speed depend upon the horse-power of the engine, stating that

$$\text{piston speed} = (660 + 6\sqrt{\text{B. H. P.}_n})\text{ ft. per min.} \qquad (1)$$

in which B. H. P._n is the normal brake horse-power.

A similar empirical formula for the revolutions per minute is

$$\text{r.p.m.} = 65 + \frac{1200}{\sqrt{\text{B.H.P.}_n}} \qquad (2)$$

For stationary and especially low-compression engines the values of equations (1) and (2) should not be exceeded. For automobile and other high-speed work the revolutions may be increased up to 1.35 times the value computed from equation (2).

I. **The First Method of Horse-power Computation** is more or less empirical in that it depends either upon the outright assumption of the mean effective pressure, or upon the computation of this factor from empirical or semi-empirical formulæ. Thus Grover bases the determination of the M. E. P. upon the compression pressure, making

$$\text{M. E. P.} = 2\,C - .01\,C^2 \tag{3}$$

where C = the pressure of compression in pounds above atmosphere. This formula is derived from an examination of a large number of indicator diagrams, but leaves out of consideration the kind of fuel or the quality of the mixture. Further, the formula gives the maximum result when the compression pressure is 100 pounds by gage. Beyond this the M. E. P. drops. The results are therefore at best approximate and in some cases absurd.

S. A. Moss, in an article entitled "Rational Methods of Gas Engine Powering," in Power, July, 1906, goes further than Grover and, in fixing upon the probable M. E. P., takes into account the size of engine as well as the kind of fuel. The cooling loss is relatively less in large than in small engines, hence the M. E. P. may be expected to increase somewhat with the size of the engine everything else remaining the same. The kind of fuel has an influence upon the power developed in any given cylinder due to the fact that the best air-fuel ratios, *i.e.*, mixtures, of the different fuels have a different heat content per cubic foot. Some of the mixtures are poorer, some richer than the average; as computed in Chapter X. Presupposing equal completeness of combustion, the richer mixtures yield the higher mean pressures, and hence the greater power. The following table shows the probable M. E. P., as computed by Moss. Only four-cycle engines are considered and the fuel is assumed to be average natural gas or illuminating gas (average heat content 87 B. T. U. per cu. ft. of mixture).

TABLE I. — VALUES OF MEAN EFFECTIVE PRESSURE FOR GAS ENGINES

Compression Pressure, Lbs. per Sq. Inch Gage	Probable Clearance as % of Displaced Volume	Mean Effective Pressures, Lbs. per Sq. Inch. Approximate Brake Horse-Power of Engine						
		5 or less	10	25	50	100	200	500
50	40	60	65	70	75			
60	35	65	70	75	80			
70	30	70	75	80	85	85	90	95
80	28	70	75	85	90	90	95	100
90	26			90	95	95	100	105
100	24			95	95	100	100	110
110	22			95	95	100	100	110
120	20					100	100	110

If some other kind of fuel than the two above specified is used, the M. E. P. of Table I should be multiplied by the proper factor from Table II following:

TABLE II. — RATIOS OF HEAT OF COMBUSTION PER CUBIC FOOT OF PERFECT MIXTURE, TO VALUE FOR AVERAGE NATURAL GAS OR MANUFACTURED ILLUMINATING GAS (87).

Oil Gas	1.00
Water Gas (uncarbureted)	1.00
Coke Oven Gas	.93
Air Gas (Siemens Producer Gas)	.79
Carbureted Water Gas	1.05
Anthracite Producer Gas	.80
Bituminous Producer Gas	.87
Blast Furnace Gas	.67
Acetylene Gas	1.28
Gasoline (liquid, vapor, or vaporized and mixed with air)	1.12
Kerosene (vaporized alone or mixed with air), and entering cylinder	.90
at about 150° F	.90

That the maximum compression pressure allowable differs for every fuel has already been pointed out in Chapter IV, where a table giving the compression pressures ordinarily employed will be found.

Having thus determined the probable M. E. P., the indicated horse-power developed at normal load by any given engine may then be computed from the formula

$$\text{I. H. P.}_n = \frac{(\text{M.E.P.})\, lan}{33000} \qquad (4)$$

where l = stroke in ft.
a = piston area in square inch.

and n = no. of explosions per minute = about $\frac{\text{r.p.m.}}{2}$ in a four-cycle machine at full load.

It will generally be found that at full load the number of explosions in a hit-and-miss engine is not quite equal to $\frac{\text{r.p.m.}}{2}$ or that in a machine governed by any of the precision methods the card given at full load is not quite the maximum card obtainable. This is done in both cases to give the engine some overload capacity, say from 10 to 20 per cent. Assuming an average overload capacity of say 15 per cent, we then have the

$$\text{I.H.P.}_n = \frac{\text{I.H.P.}_m}{1.15} \tag{5}$$

where I. H. P.$_n$ = normal or full load I. H. P., and
I. H. P.$_m$ = maximum I. H. P. obtainable.

Now at full load the mechanical efficiency of an average engine may be assumed to be 80 per cent, so that the normal brake horse-power will be

$$\text{B.H.P.}_n = .8\ \text{I.H.P.}_n = \frac{.8}{1.15}\ \text{I.H.P.}_m = .695\ \text{I.H.P.}_m \tag{6}$$

or

$$\text{I.H.P.}_m = 1.44\ \text{B.H.P.}_n \tag{7}$$

In the case of an engine to be constructed, B. H. P.$_n$ is usually specified, hence I. H. P.$_n$ can be found from equation (6). Substituting this in equation (4) leaves as the unknown factors in this equation the factors l, a, and n. The latter depends upon the revolutions, which may also be specified or determined from equation (2). This leaves only l and a to be determined.

It is next necessary to find some relation between cylinder diameter and stroke. An examination of existing engines shows that practice in this regard is pretty well settled. High-speed engines show a ratio of $\frac{\text{stroke}}{\text{diameter}}$ = 1.1 to 1.3. In medium speed engines the ratio varies from about 1.2 to 1.6, and large machines vary between 1.5 and 2.0.

To determine the cylinder dimensions, let d be the diameter of the cylinder in feet and let

$$l = x\,d \qquad (8)$$

Then equation (4) may be rewritten for *maximum* power,

$$\text{I.H.P.}_m = \frac{(\text{M. E. P.}) \times xd \times .785\, d^2 \times 144 \times n}{33000}$$

$$= \frac{(\text{M. E. P.})x d^3 n}{300} \qquad (9)$$

From this the cylinder diameter is

$$d = \sqrt[3]{\frac{300\ \text{I.H.P.}_m}{(\text{M.E.P.})\,xn}}\ \text{ft.} \qquad (10)$$

EXAMPLES. — 1. To show the method of using these equations, suppose it is desired to determine the required cylinder dimensions for a 100 B. H. P., four-cycle throttling engine, using anthracite producer gas.

From equation (2), the number of revolutions should be about

$$\text{r.p.m.} = 65 + \frac{1200}{\sqrt{100}} = 185$$

From the table of Chapter IV, a compression pressure of 120 pounds by gage is about normal for the fuel used. From Tables I and II we should expect a mean effective pressure for this case of

$$\text{M. E. P.} = 100 \times .80 = 80 \text{ pounds.}$$

From equation (7), $\text{I. H. P.}_m = 1.44 \text{ B. H. P.}_n = 1.44 \times 100 = 144$

The engine being of the throttling type,

$$n = \frac{\text{r.p.m.}}{2} = \frac{185}{2} = 92.5$$

and x may be assumed equal to 1.6.

Substituting these values in equation (10), we finally have

$$\text{cyl. dia. } d = \sqrt[3]{\frac{300 \times 144}{80 \times 1.6 \times 92.5}} = \sqrt[3]{3.65} = 1.54 \text{ ft.}$$

$$= \text{Appr. } 18\tfrac{1}{2}''$$

Stroke $= 1.6 \times 18.5'' = $ app. $29\tfrac{5}{8}''$

and r.p.m. $= 185$

2. A four-cycle gasoline engine has a cylinder diameter of 6″, a stroke of 8½″, and the number of revolutions is 250 per minute. It is governed by hit and miss. What is its probable brake horse-power at normal load?

The M. E. P. at normal I. H. P. from Tables I and II is about

$$1.12 \times 75 = 84 \text{ pounds}$$

assuming that the compression pressure is about 70 pounds by gage. For the maximum indicated horse-power, the number of explosions n may be assumed to be equal to $\frac{\text{r.p.m.}}{2} = \frac{250}{2} = 125.$

Substituting in equation (4)

$$\text{I. H. P.}_m = 1.15\,\text{I. H. P.}_n = 144\,\text{B. H. P.}_n = \frac{84 \times \frac{8.5}{12} \times 785 \times 6^2 \times 125}{33000}$$
$$= 6.35$$

From which $$\text{B. H. P.}_n = \frac{6.35}{1.44} = \text{appr. } 4.4$$

II. The Second Method of determining mean effective pressure follows steam engine practice, and is developed in detail by Lucke in his Gas Engine Design. The method consists of drawing first a standard air reference diagram. From this the mean effective pressure is determined by multiplying the mean effective pressure of the standard diagram by a card or diagram factor depending upon the kind of fuel used and the compression carried. Thus fundamentally this method does not differ greatly from that of Moss outlined above.

The standard air reference diagram is obtained by working a pound of air, which is assumed to receive as much heat as a pound of the proper fuel-air mixture to be used in the real engine would contain, through the cycle.

The following table shows the card factors as determined by Lucke from a large number of constructed engines:

Kind of Fuel and Method of Use	Range of Compression by Gage	Card Factor %
Kerosene, when previously vaporized	45–75	30–40
Kerosene, injected on hot bulb, may be as low as	....	20
Gasoline, used in carbureter requiring a vacuum, depending upon the extent of the vacuum	....	25–40
Gasoline, with but little initial vacuum	80–130	50–30
Producer gas	100–160	56–40
Coal gas	Av. 80	Av. 45
Blast furnace gas	130–180	48–30
Natural gas	90–140	52–40

NOTE. — Card factors for two-cycle engines may be taken as .8 that for four-cycle machines.

The mean effective pressure of the standard air reference diagram, of course increases continuously with the compression. The above table, however, shows that the card factor decreases as the compression increases for all fuels except kerosene. This accounts for the fact that in Moss's Table I the M. E. P. does not increase beyond certain compressions. The thermal and mechanical reasons underlying this fact have been explained in Chap. IV.

The table of card factors shows a large range of variation in the different factors and emphasizes the point made by Lucke that in determining cylinder sizes a great deal must be left to the personal experience and judgment of the designer.

III. Third Method of Determining Cylinder Dimensions. It must be obvious from what has been said that it is not at all easy to predict with fair accuracy the mean effective pressure an engine may be expected to realize. For that reason a method of determining the cylinder dimensions of a proposed engine, or the power of an existing engine, without first determining the M. E. P., should be welcome.

The method, developed by Güldner,* is based upon two well-known facts:

1. The power developed by any gas engine depends directly upon the volume of the mixture it can handle in unit time.

2. The power also depends upon the thermal efficiency with which the engine can handle this volume of mixture.

Regarding the first point, the charge volume in unit time

* Güldner, Entwerfen & Berechnen der Verbrennungsmotoren.

involves cylinder diameter, stroke, and number of revolutions. Two of these factors can usually be fixed upon, which determines the third. The computation of the charge volume for a given time involves the assumption of a value for the volumetric efficiency of the suction stroke. There is now so much experimental data regarding this point that no great error can be made. A table of volumetric efficiencies E_v for various types of engines is given on page 86.

The same is true of the thermal efficiency and fuel characteristics. A large number of tests on various types of machines and different fuels enables us to-day to forecast with a fair degree of accuracy what thermal efficiency may be expected from a given engine when its approximate power and the kind of fuel used are known.

The following derivation of equations for values of diameter, stroke and revolutions per minute is due to Güldner,* as is also the appended table. All metric measurements and values have been transformed to English units:

Let B. H. P.$_n$ or N_n = nominal brake horse-power.

n = R. P. M.

d = piston diameter in feet.

l = stroke in feet.

$V_h = .785\ d^2l$ = piston displacement per stroke in cu. ft.

$V = E_vV_h$ = actual volume of mixture in cu. ft. per suction stroke, barometer 28.95″, temperature 59 degrees Fahrenheit.

$E_v = \dfrac{V}{V_h}$ = volumetric efficiency of suction stroke.

L = the volume of air in cu. ft. required for 1 cu. ft. of gas fuel, or 1 pound of liquid fuel, under most favorable practical conditions. This is not the theoretical quantity, but some excess quantity as seems best for the particular fuel.

* The article may be found either in the Zeitschrift des Vereines deutscher Ingenieure, April 26, 1902, or in his book, Entwerfen und Berechnen der Verbrennungsmotoren, pp. 213–215.

L_h = the resulting actual quantity of air in cu. ft. for one explosion, for nominal horse-power.

C_s = the quantity of fuel used per hour, for gases in cu. ft., for liquids in pounds, at nominal horse-power.

C = the same per horse-power hour.

C_h = the same per explosion.

H = the lower heating value of the fuel, for gases per cu. ft., for liquids, per pound, in B. T. U.

$$e = \frac{N_n \times 33000 \times 60}{778\, C_s H} = \frac{2545\, N_n}{C_s H} = \left\{ \begin{array}{l} \text{thermal efficiency at the} \\ \text{brake} = \text{economic efficiency.} \end{array} \right.$$

From the above we can derive for four-cycle engines

$$C_s = \frac{N_n \times 33000 \times 60}{778\, He} = \frac{2545\, N_n}{He} \tag{11}$$

$$C_h = \frac{2\, C_s}{60\, n} = \frac{2 \times 2545\, N_n}{60\, nHe} = \frac{84.8\, N_n}{nHe} \tag{12}$$

$$L_h = \frac{C_s L}{30\, n} = \frac{2545\, N_n L}{30\, nHe} = \frac{84.8\, N_n L}{nHe} \tag{13}$$

For two-cycle engines equations (12) and (13) should be divided by 2, because there is a charging stroke for every revolution.

ENGINES FOR GAS FUEL

The actual charge taken in by the engine during one suction stroke must be

$$V = C_h + L_h$$

This volume requires a piston displacement of

$$V_h = .785\, d^2 l = \frac{C_h + L_h}{E_v} = \frac{84.8\, N_n + 84.8\, N_n L}{nHeE_v}$$

$$= \frac{84.8\, N_n (1 + L)}{nHeE_v} \text{ cubic feet} \tag{14}$$

Solving (14) for d, l and n in turn, we have

$$d = \sqrt{\frac{108\, N_n\,(1 + L)}{enHlE_v}}\ \text{feet} \tag{15}$$

$$l = \frac{108\, N_n\,(1 + L)}{enHd^2E_v}\ \text{feet} \tag{16}$$

$$n = \frac{108\, N_n\,(1 + L)}{elHd^2E_v}\ \text{feet} \tag{17}$$

ENGINES FOR LIQUID FUEL

In this case the fuel is introduced either in liquid form or in the shape of vapor. In either case the volume ratios of fuel to air are very much smaller than they are in the case of gaseous fuels. Take for instance the case of alcohol, a fuel of low heating value. Here we find that the vapor does not form theoretically more than 4 per cent of the volume of the charge. In reality it is even less than this on account of the excess air used. In view of this fact we may put for these engines

$$V_k = .785\ d^2l = \frac{L_k}{E_v} = \frac{84.8\ N_nL}{nHeE_v}\ \text{cubic feet} \tag{18}$$

Solving this again for d, l and n we obtain for engines using liquid fuel:

$$d = \sqrt{\frac{108\ N_nL}{enHlE_v}}\ \text{ft.} \tag{19}$$

$$l = \frac{108\ N_nL}{enHd^2E_v}\ \text{ft.} \tag{20}$$

and

$$n = \frac{108\ N_nL}{elHd^2E_v} \tag{21}$$

In these equations the quantities to which certain values must be assigned are L, e and E'_v. As stated before, there is now so much practical data at hand that the proper determination of these should cause no difficulty. For values of E_v see Table, p. 86. The table on the opposite page gives values of L, C and e, as found for various fuels.

Since the nominal brake horse-power N_n is 18 to 20 per cent below the maximum capacity of engines in ordinary cases, it is well to assume an excess of air of about 30 per cent at the outset, in the case of machines using gas. In the case of liquid fuel

	Lower Heating Value (H) B.T.U. per		Air required, Cubic Feet				Consumption of Fuel (C) per Brake Horse-Power Hour, and Thermal Brake Efficiency (e), when N_n =														
			Theoretically (L_0) per		Actually (L) per		5 B.H.P.			10 B.H.P.			25 B.H.P.			50 B.H.P.			100 B.H.P. and over		
							(C)		(e)	(C)		(e)	(C)		(e)	(C)		(e)	(C)		(e)
	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lb.	Cu. ft.	Lbs.		Cu. ft.	Lbs.		Cu. ft.	Lbs.		Cu. ft.	Lbs.		Cu. ft.	Lbs.	
I. Illuminating gas																					
a. Lean	505	...	5.5	...	7.5	...	24.8	...	.20	22.2	...	.22	20.4	...	.24	19.1	...	.26	18.5	...	.27
b. Ordinary	560	...	to	...	to	...	22.2	...	.20	20.1	...	.22	18.3	...	.24	16.9	...	.26	16.5	...	.27
	618	...		...		...	20.4	...	.20	18.3	...	.22	16.9	...	.24	15.5	...	.26	15.1	...	.27
c. Rich	675	...	6.5	...	10.0	...	18.7	...	.20	16.7	...	.22	15.5	...	.24	14.1	...	.26	13.7	...	.27
II. Generator gas																					
(a) On basis of anthracite	...	13400	...	...	...	...	...	1.65	.11	...	1.43	.13	...	1.25	.15	...	1.10	.17	...	1.03	.18
(b) Anthracite gas	141	...	.85	...	1.1	...	106.0	...	.17	95.3	...	.19	84.8	...	.21	77.6	...	.23	73.2	...	.24
(c) On basis of coke	...	12950	to	...	to	...	...	1.76	.11	...	1.49	.13	...	1.29	.15	...	1.14	.17	...	1.07	.18
(d) Coke gas	129	...	1.00	...	1.4	...	116.0	...	.17	102.0	...	.19	91.8	...	.21	84.8	...	.23	81.3	...	.24
III. Blast furnace gas	106	...	.75	...	1.0-1.2	...	...	...	...	130.0	...	.18	116.0	...	.20	102.0	...	.22	98.8	...	.24
IV. Coke oven gas	505	...	5.3	...	7.0	...	...	...	...	35.3	...	.17	30.0	...	.19	26.4	...	.21	24.7	...	.23
V. Kerosene	...	18900	...	185.0	...	257–353	...	1.25	.11	...	1.10	.12	...	.99	.13	...	...	...	...	...	...
VI. Crude oil (Diesel)	...	18000	...	176	...	288–323	...	.55	.25	...	.527	.26	...	.506	.27	...	.462	.30	...	.440	.315
VII. Gasoline	...	19800	...	176	...	240–323	...	.66	.19	...	.615	.21	...	.550	.23	...	...	...	...	...	...
VIII. Alcohol (90% vol.)	...	10300	...	96.5	...	128–193	...	1.10	.22	...	.99	.24	...	.92	.26	...	...	...	...	...	...

engines a greater excess, 50 to 60 per cent, should be used, because these fuels are usually high in heating value, and if any economical degree of compression is to be employed, the fuel mixture should be rather lean. Besides it is not easy to produce a uniform fuel mixture in the case of liquid fuels, and an excess of air would help to overcome this difficulty. These excess air allowances have been made in the table. The values of C and e are for the various fuels and for various sizes of engines, as determined from existing data. The fuel consumed by heating apparatus and ignition flames is not included in C. In the case of engines using generator gas it is supposed that suction generators are employed requiring no fuel for a separate boiler.

To see how the results obtained by Güldner's method agree with those obtained by Moss, the same examples will be taken.

Example 1. — Engine to be 100 B. H. P., throttling governor, fuel to be anthracite producer gas. Determine the cylinder dimensions, assuming as before that the number of revolutions is 185 per minute.

For this case any one of the formulæ 15 to 17 may be used. Taking the first, we have

$$d = \sqrt{\frac{108\,N_n\,(1 + L)}{enHlE_v}} \text{ feet}$$

in which N_n = 100 B. H. P.

L = air used per cu. ft. of the fuel gas = say 1.3 for anthracite gas from the table.

e = thermal efficiency that may be expected = .24 from the table.

n = 185.

H = heating value of fuel = 141 B. T. U. from the table.

l = Stroke = $x\,d$ = assumed as 1.6 d in previous case.

E_v = volumetric efficiency = say .90 from table p. 86.

Then $$d^2 = \frac{108 \times 100 \times 2.3}{.24 \times .185 \times 141 \times 1.6\,d \times .90}$$

or $$d^3 = \frac{108 \times 100 \times 2.3}{.24 \times 185 \times 141 \times 1.6 \times .90} = 2.77$$

from which $d = 1.42$ ft. = appr. 17″, and

$$l = 1.6\,d = 1.6 \times 17'' = \text{appr. } 27\tfrac{1}{2}''.$$

This method, therefore, apparently calls for a smaller engine.

EXAMPLE 2.— Gasoline engine, cyl. diam. 6″, stroke $8\frac{1}{2}''$, r.p.m. 250. Hit-and-miss governor. Determine probable B. H. P.

For this, equation 21 may be rewritten

$$N_n = \frac{nelHd^2E_v}{108\,L}$$

in which n = 250 r.p.m.

e = thermal efficiency = .19 from table.

l = stroke = $8\frac{1}{2}''$ = .71 ft.

H = heating value of fuel = 19800, from table.

d = 6″ = .5 ft., d^2 = .25.

E_v = .75, from p. 86, since this engine is likely to have an automatic inlet valve.

L = 250, for a good rich mixture.

Then

$$N_n = \frac{250 \times .19 \times .71 \times 19800 \times .25 \times .75}{108 \times 250} = 4.65 \text{ B.H.P.}$$

The previous example gave 4.4 B. H. P. so that the agreement in this case is satisfactory.

IV. For determining the power rating of automobile engines, several more or less empirical formulæ are in use, two of which will be cited.

The Association of Automobile Manufacturers has fixed upon the following expression for four-cycle engines:

$$\text{B.H.P.} = \frac{d^2N}{2.5}$$

where d = diameter of cylinder in inches, and

N = number of cylinders.

This is based on a speed of 1000 r.p.m. In this formula, however, the factor 2.5 results from the contraction of several other factors more or less arbitrarily assumed, and hence as far as design is concerned the formula is of little or no use.

$$\text{B.H.P.} = \frac{d^2 l N n}{14000}\left(.48 + \frac{1}{10\,Cl}\right)$$

where d = cyl. diam. in inches.
l = stroke in inches.
N = no. of cylinders.
n = r.p.m., and
Cl = clearance expressed in terms of piston displacement.

The following example shows the application of the methods discussed to the determination of the power of an automobile machine.

EXAMPLE. — Cylinder diameter 4″, stroke 4½″, r.p.m. 1000, no. of cylinders 4. Determine the probable B. H. P.

I. By determination of M. E. P.

Assume compression carried is 80 pounds, which requires about 28 per cent clearance. From Moss's Tables

$$\text{M. E. P.} = 75 \times 1.12 = 84 \text{ pounds.}$$

From equation (4) therefore

$$\text{I. H. P.}_n = \frac{84 \times \frac{4.5}{12} \times .785 \times 4^2 \times 500}{33000} = 6 \text{ H. P.}$$

Since the mechanical efficiency is in the neighborhood of .80,

$$\text{B. H. P.} = .8 \times 6 = 4.8$$

and for four cylinders

$$\text{Total B. H. P.} = 4 \times 4.8 = 19.2$$

II. By Güldner's Method:

$$N_n = \frac{nelHd^2E_v}{108\,L}$$

$$= \frac{1000 \times .20 \times \frac{4.5}{12} \times 19800 \times \left(\frac{4}{12}\right)^2 \times .75}{108 \times 250} = 4.6$$

and total B. H. P. = $4 \times 4.6 = 18.4$

III. By Association Formula:

$$\text{B. H. P.} = \frac{4^2 \times 4}{2.5} = \frac{64}{2.5} = 25.6$$

IV. By Rice's Formula:

$$\text{B.H.P.} = \frac{4^2 \times 4.5 \times 4 \times 1000}{14000}\left(.48 + \frac{1}{10 \times .28}\right) = 17.3$$

Of the above results, those obtained by methods I, II, and IV agree very well. Method III gives a result considerably higher, but in view of the fact that the choice of the factor 2.5 is somewhat arbitrary, this discrepancy does not mean much. If the factor had been taken equal to three, for instance, as is sometimes done, the result would be brought down to 21.3 horse-power which is not so far different from the rest.

CHAPTER XVI

METHODS OF TESTING GAS ENGINES

THE actual testing of gas engines, that is, the determination of indicated and brake horse-power, speed, fuel consumption, etc., does not differ materially from the methods long become standard in steam engine practice. But the calculations involved in the proper working up of the data obtained are enough different to have caused engineering societies at home and abroad to set up rules and regulations, so-called codes, for testing internal combustion engines.

Thus in 1898 the American Society of Mechanical Engineers appointed a committee for this purpose which rendered a final report in an Appendix to the Steam Engine Code in 1901. The British Institution of Civil Engineers followed suit, the preliminary and final reports being rendered in March and December, 1905, respectively. Finally the Verein Deutscher Ingenieure adopted a code for the testing of gas engines and gas producers, which was published in the *Zeitschrift* for November 24, 1906. The latter code was translated nearly entirely by Mr. F. E. Junge and will be found in Power for February, 1907.

Of these codes the American and German give specific rules for testing, the latter paying particular attention to acceptance tests of both engines and gas producers. The British report concerns itself mainly with establishing efficiency standards.

In describing the methods followed in the testing of gas engines, it was thought best to give the American code in its entirety, supplementing it by most of the references to the steam engine code contained in its original printed form. This is followed by a translation of nearly the entire German code, modified where necessary to suit the American conditions. This code is also given because it not only supplements the American code in some important details, but on account of the rules given for the testing of gas producers.

RULES FOR CONDUCTING TESTS OF GAS AND OIL ENGINES. CODE OF 1901

I. Objects of the Tests. — At the outset the specific object of the test should be ascertained, whether it be to determine the fulfilment of a contract guarantee, to ascertain the highest economy obtainable, to find the working economy and the defects as they exist, to ascertain the performance under special conditions, or to determine the effect of changes in the conditions; and the test should be arranged accordingly.

Much depends upon the local conditions as to what preparations should be made for a test, and this must be determined largely by the good sense, tact, judgment, and ingenuity of the expert undertaking it, keeping in mind the main issue, which is to obtain accurate and reliable data. In deciding questions of contract, a clear understanding in regard to the methods of test should be agreed upon beforehand with all parties, unless these are distinctly provided for in the contract.

II. General Condition of the Engine. — Examine the engine, and make notes of its general condition, and any points of design, construction, or operation which bear on the objects in view. Make a special examination of all the valves by inspecting the seats and bearing surfaces, and note their condition, and see if the piston rings are gas-tight.

If the trial is made to determine the highest efficiency, and the examination shows evidence of leakage, the valves and piston rings, etc., should be made tight, and all parts of the engine put in the best possible working condition before starting on the test.

III. Dimensions, etc. — Take the dimensions of the cylinder, or cylinders, whether already known or not; this should be done when they are hot, and in working order. If they are slightly worn the average diameter should be determined. Measure, also, the compression space or clearance volume, which should be done, if practicable, by filling the spaces with water previously measured, the proper correction being made for the temperature.

IV. Fuel. — Decide upon the gas or oil to be used, and if the trial is to be made for maximum efficiency, the fuel should be the best of its class that can readily be obtained, or one that shows the highest calorific power.

V. Calibration of Instruments used in the Tests. — All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards. Apparatus liable to change or to become broken during the tests, such as gages, indicator springs, and thermometers, should be calibrated both before and after the experiments. The accuracy of all scales should be verified by standard weights. In the case of gas or water meters, special attention should be given to their calibration, both before and after the trial, and at the same rate of flow and pressure as exists during the trial.

(*a*) Gages. — For pressures above the atmosphere, one of the most convenient, and at the same time reliable, standards is the dead-weight testing apparatus which is manufactured by many of the prominent gage makers. It consists of a vertical plunger nicely fitted into a cylinder containing oil or glycerine, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights may be placed, and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger at any time, divided by the area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison for pressures is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever varying zero point; that the mercury is pure, and that the proper correction is made for any difference of temperature that exists, compared with the temperature at which the instrument was graduated.

For pressure below the atmosphere, an air pump or some other means of producing a vacuum is required, and reference must be made to a mercury gage. Such a gage may be a U-tube having a length of 30 inches or so, with both arms properly filled with pure mercury.

(*b*) Thermometers. — Standard thermometers are those which indicate 212 degrees Fahrenheit in steam escaping from boiling water at the normal barometrical pressure of 29.92 inches, the whole stem up to the 212-degree point being surrounded by the steam; and which indicate 32 degrees Fahrenheit in melting

ice, the stem being likewise completely immersed to the 32-degree point; and which are calibrated for points between and beyond these two references points. We recommend, for temperatures between 212 degrees and 400 degrees Fahrenheit, that the comparison of the thermometer be made with the temperature given in Regnault's Steam Tables, the method required being to place it in a mercury well surrounded by saturated steam under sufficient pressure to give the right temperature. The pressure should be accurately determined as pointed out in the above section (*a*), and the thermometer should be immersed to the same extent as it is under its working condition.

Thermometers in practice are seldom used with the stems fully immersed; consequently, when they are compared with the standard, the comparison should be made under like conditions, whatever those happen to be.

If pyrometers of any kind are used, they should be compared with a mercury thermometer within its range, and if extreme accuracy is required with an air thermometer, or a standard based thereon, at higher points, care being taken that the medium surrounding the pyrometer, be it air or liquid, is of the same uniform temperature as that surrounding the standard.

(*c*) Indicator Springs. — The indicator springs should be calibrated with the indicator in as nearly as possible the same condition as to temperature as exists during the trial. This temperature can usually be estimated in any particular case. A simple way of heating the indicator is to subject it to a steam pressure just before calibration. Compressed air, or compressed carbonic acid gas, are suitable for the actual work of calibration. These gases should be used in preference to steam, so as to bring the conditions as near as possible to those which obtain when the indicators are in actual use. When compressed carbonic acid gas is used, and trouble arises from the clogging of the escape valves with ice, the pipe between the valve and the gas tank should be heated. With both air and carbonic acid gas, the pipes leading to the indicator should also be heated if it is found that they are below the required temperature. The springs may be calibrated for this class of engines under a constant pressure, if desired, and the most satisfactory method is to cover the whole range of pressure through which the indicator acts; first, by grad-

ually increasing it from the lowest to the highest point, and then gradually reducing it from the highest to the lowest point, in the manner which has heretofore been widely followed by indicator makers; a mean of the results should be taken. The calibration should be made for at least five points, two of these being for the pressures corresponding to the maximum and minimum pressures, and three for intermediate points equally distant.

The standard of comparison recommended is the dead weight testing apparatus, a mercury column, or a steam gage, which has been proved correct by reference to either of these standards.

When the scale of the spring determined by calibration is found to vary from the nominal scale with substantial uniformity, it is usually sufficiently accurate to take the arithmetical mean of the scales found at the different pressures tried. When, however, the scale varies considerably at the different points, and absolute accuracy is desired, the method to be pursued is as follows: Select a sample diagram and divide it into a number of parts by means of lines parallel to the atmospheric line, the number of lines being equal to and corresponding with the number of points at which the calibration of the spring is made. Take the mean scale of the spring for each division and multiply it by the area of the diagram inclosed between two contiguous lines. Add all the products together and divide by the area of the whole diagram; the result will be the average scale of the spring to be used. If the sample diagram selected is a fair representative of the entire set of diagrams taken during the test, this average scale can be applied to the whole. If not, a sufficient number of samples of diagrams representing the various conditions can be selected, and the average scale determined by a similar method for each, and thereby the average for the whole run.

(*d*) Gas Meters. — A meter used for measuring gas for a gas engine should be calibrated by referring its readings to the displacement of a gasometer of known volume, by comparing it with a standard gas meter of known error, or by passing air through the meter from a tank in which air under pressure is stored. If the latter method is adopted, it is necessary to observe the pressure of the air in the tank and its temperature, both at the tank and at the meter, and this should be done at uniform intervals during the progress of the calibration. The amount of

air passing through the meter is computed from the volume of the tank and the observed temperatures and pressures.

The volume of the gas thus ascertained should be reduced to the equivalent at a given temperature and atmospheric pressure, corrected for the effect of moisture in the gas, which is ordinarily at the saturation point or nearly so. We recommend that a standard be adopted for gas-engine work, the same as that used in photometry, namely, the equivalent volume of the gas when saturated with moisture at the normal atmospheric pressure at a temperature of 60 degrees Fahrenheit. In order to reduce the reading of the volume containing moist gas at any other temperature to this standard, multiply by the factor

$$\frac{459.4 + 60}{459.4 + t} \times \frac{b - (29.92 - s)}{29.4},$$

in which b is the height of the barometer in inches at 32 degrees Fahrenheit, t the temperature of the gas at the meter in degrees Fahrenheit, and s the vacuum in inches of mercury corresponding to the temperature of t obtained from steam tables.

(*e*) Water Meters. — A good method of calibrating a water meter is the following, reference being made to Fig. 16–1.

Two tees A and B are placed in the feed pipe, and between them two valves C and D. The meter is connected between the outlets of the tees A and B. The valves E and F are placed one on each side of the meter. When the meter is running, the valves E and F are opened, and the valves C and D are closed. Should an accident happen to the meter during the test, the valves E and F may be closed, and the valves C and D opened, so as to allow the feed water to flow directly to the point of use. A small bleeder G is

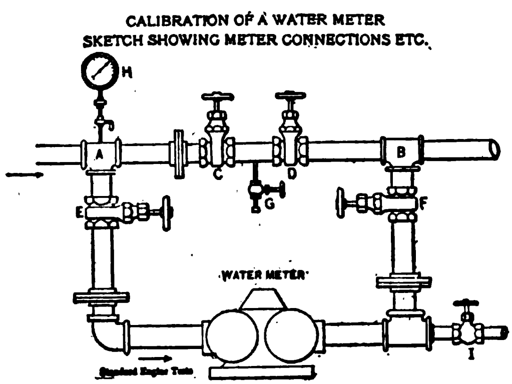


Fig. 16–1.

opened when the valves C and D are closed, in order to make sure that there is no leakage. A gage is attached at H. When the meter is tested, the valves C, D, and F are closed, and the valves E and I are opened. The water flows from the valve I to a tank placed on weighing scales. In testing the meter the rate of flow should be the same as that on test, and the water leaving the meter is throttled at the valve I until the pressure shown by the gage H is the same as that indicated when the meter is running under the normal conditions. The piping leading from the valve I to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be swung readily into the tank or away from it. After the desired pressure and rate of flow have been secured, the end of the pipe is swung into the tank the instant that the pointer of the meter is opposite some graduation mark on the dial, and the water continues to empty into the tank. The tests should be made by starting and stopping at the same graduation mark on the meter dial, and continued until at least 10 or 20 cubic feet are discharged for one test. The water collected in the tank is then weighed.

The water passing the meter should always be under pressure in order that any air in the meter may be discharged through the vents provided for this purpose. Care should be taken that there is no air contained in the water. The meter should be tested both before and after the engine trial, and several tests be made of the meter in each case in order to obtain confirmative results. It is well to make preliminary tests to determine whether the meter works satisfactorily before connecting it up for an engine trial. The results should agree with each other for two widely different rates of flow.

VI. Duration of Test. — The duration of a test should depend upon its character and the objects in view, and in any case the test should be continued until the consecutive readings of the rates at which oil or gas is consumed, taken at say half-hourly intervals, become uniform and thus verify each other. If the object is to determine the working economy, and the period of time during which the engine is usually in motion is some part of twenty-four hours, the duration of the test should be fixed for this number of hours. If the engine is one using coal for generating gas, the test should cover a long enough period to determine

with accuracy the coal used in the gas producer; such a test should be of at least twenty-four hours' duration, and in most cases it should extend over several days.

VII. Starting and Stopping a Test. — In a test for determining the maximum economy of an engine, it should first be run a sufficient time to bring all the conditions to a normal and constant state. Then the regular observations of the test should begin, and continue for the allotted time.

If a test is made to determine the performance under working conditions, the test should begin as soon as the regular preparations have been made for starting the engine in practical work, and the measurements should then commence and be continued until the close of the period covered by the day's work.

VIII. Measurement of Fuel. — If the fuel used is coal furnished to a gas producer, the same methods apply for determining the consumption as are used in steam boiler tests.

If the fuel used be gas, the only practical method of measurement is the use of a meter through which the gas is passed. Gas bags should be placed between the meter and the engine to diminish the variation of pressure, and these should be of a size proportionate to the quantity used. When a meter is employed to measure the air used by an engine, a receiver with a flexible diaphragm should be placed between the engine and the meter. The temperature and pressure of the gas should be measured, as also the barometric pressure and temperature of the atmosphere, and the quantity of gas should be determined by reference to the calibration of the meter, taking into account the temperature and pressure of the gas. (See Section V (*d*)).

If the fuel is oil, this can be drawn from a tank which is filled to the original level at the end of the test, the amount of oil required for so doing being weighed; or, for a small engine, the oil may be drawn from a calibrated vessel such as a vertical pipe.

In an engine using an igniting flame the gas or oil required for it should be included in that of the main supply, but the amount so used should be stated separately, if possible.

IX. Measurement of Heat-Units Consumed by the Engine. — The number of heat units used is found by multiplying the number of pounds of coal or oil or the cubic feet of gas consumed, by the total heat of combustion of the fuel as determined by a cal-

orimeter test. In determining the total heat of combustion no deduction is made for the latent heat of the water vapor in the products of combustion. There is a difference of opinion on the propriety of using this higher heating value, and for purposes of comparison care must be taken to note whether this or the lower value has been used. The calorimeter recommended for determining the heat of combustion is the Mahler, for solid fuels or oil, or the Junker for gases, or some form of calorimeter known to be equally reliable. (See Chapter VI, or Poole on "The Calorific Power of Fuels.")

It is sometimes desirable, also, to have a complete chemical analysis of the oil or gas. The total heat of combustion may be computed, if desired, from the results of the analysis, and should agree well with the calorimeter values.

In using the gas calorimeter, which involves the determination of the volume instead of the weight of the gas, it is important that this should be reduced to the same temperature as that corresponding to the conditions of the engine trial. The formula to be used for making the reduction is that already given in Section V (*d*).

For the purpose of making the calorimeter test, if the fuel used is coal for generating gas in a producer, or oil, samples should be taken at the time of the engine trial, and carefully preserved for subsequent determination. If gas is used, it is better to have a gas calorimeter on the spot, samples taken, and the calorimeter test made while the trial is going on.

X. Measurement of Jacket Water to Cylinder or Cylinders. — The jacket water may be measured by passing it through a water meter or allowing it to flow from a measuring tank before entering the jacket, or by collecting it in tanks on its discharge.

XI. Indicated Horse-Power. — The directions given for determining the indicated horse-power for steam engines apply in all respects to internal combustion engines.

The indicated horse-power should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes this necessary. With variable loads, such as those of engines driving generators for electric railroad work, and of rubber-grinding and rolling mill engines, the diagrams cannot be

taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of ten seconds until two minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a "variation diagram" should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

AUTHOR'S NOTE. — When the engine is governed by the hit-and-miss principle the diagrams taken on one card should in any case cover the series of consecutive explosions, and the mean diagram should be used as the basis of calculations.

The most satisfactory driving rig for indicating seems to be some form of well-made pantagraph, with driving cord or fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal lengths, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

To test the accuracy of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equidistant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a

reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the piston has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length of the diagram is the proportion desired.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely — so as to get absolute freedom of motion — than to have a mechanically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams. No diagrams should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion or compression lines is a sure indication that the piston or some part of the mechanism sticks; and when this state of things is revealed the indicator should not be trusted, but the cause should be ascertained and a suitable remedy applied. An indicator which is free when subjected to a steady pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger, as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to the pressure, the movement should appear smooth to the sense of feeling.

The pipe connections for indicating gas and oil engines should be removed as far as possible from the ports and ignition devices, and made preferably in the cylinder head. The pipes should be

as short and direct as possible. Avoid the use of long pipes, otherwise explosions of the gas in these connections may occur.

Ordinary indicators suitable for indicating steam engines are much too lightly constructed for gas and oil engines. The pencil mechanism, especially the pencil arm, needs to be very strong to prevent injury by the sudden impact at the instant of the explosion; a special gas-engine indicator is required for satisfactory work, with a small piston and a small spring.

See Chapter I for the description of various indicators.

XII. Brake Horse-Power. — The determination of the brake horse-power, which is very desirable, is the same for internal combustion as for steam engines.

This term applies to the power delivered from the fly-wheel shaft to the engine. It is the power absorbed by a friction brake applied to the rim of the wheel, or to the shaft. A form of brake is preferred that is self-adjusting to a certain extent, so that it will, of itself, tend to maintain a constant resistance at the rim of the wheel. One of the simplest brakes for comparatively small engines, which may be made to embody this principle, consists of a cotton or hemp rope, or a number of ropes, encircling the wheel, arranged with weighing scales or other means for showing the strain. An ordinary band brake may also be constructed so as to embody the principle. The wheel should be provided with interior flanges for holding water used for keeping the rim cool.

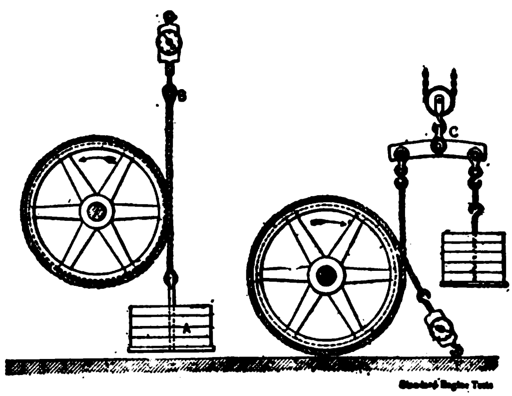


FIG. 16-2.

A self-adjusting rope brake is illustrated in Fig. 16-2, where it will be seen that, if the friction at the rim of the wheel increases, it will lift the weight A, which action will diminsh the tension in the end B of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the

ordinary construction. Where space below the wheel is limited, a cross bar, C, supported by a chain tackle exactly at its center point, may be used as shown in Fig. 16–2, thereby causing the action of the weight on the brake to be upward. A safety stop should be used with either form, to prevent the weights being accidentally raised more than a certain amount.

The water-friction brake is specially adapted for high speeds and has the advantage of being self-cooling. The Alden brake is also self-cooling and is capable of fine adjustment.

A water-friction brake is shown in Fig. 16–3. It consists of two circular discs, A and B, attached to the shaft C, and revolving in a case, E, between fixed planes. The space between the discs and the planes is supplied with running water, which enters at D and escapes at the cocks F, G, and H. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the ring of water, the greater amount of power absorbed. By suitably adjusting the amount of water entering and leaving any desired power can be obtained. Water-friction brakes have been used successfully at speeds of over 20,000 revolutions per minute.

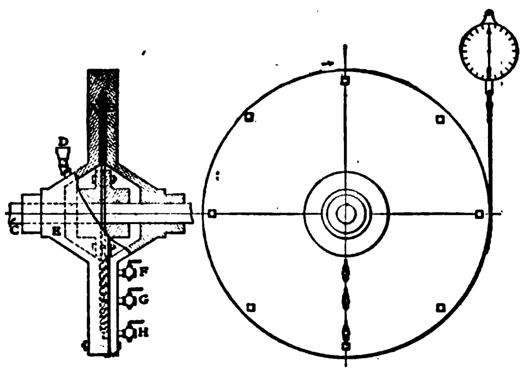


FIG. 16–3.

For methods of computing brake horse-power see Chapter I.

XIII. Speed. — There are several reliable methods of ascertaining the speed, or the number of revolutions of the engine crank shaft per minute. The simplest is the familiar method of counting a number of turns for a period of one minute with the eye fixed on the second hand of a time piece. Another is the use of a counter held for a minute or a number of minutes against

the end of the main shaft. Another is the use of a reliable tachometer held likewise against the end of the shaft. The most reliable method, and the one we recommend, is the use of a continuous recording engine register or counter, taking the total reading each time that the general test data are recorded, and computing the revolutions per minute corresponding to the difference in the readings of the instrument. When the speed is above 250 revolutions per minute, it is almost impossible to make a satisfactory counting of the revolutions without the use of some kind of mechanical counter.

The determination of variation of speed during a single revolution, or the effect of the fluctuation due to sudden changes of the load, is also desirable, especially in engines driving electric generators used for lighting purposes. There is at present no recognized standard method of making such determinations, and if such are desired, the method employed may be devised by the person making the test and described in detail in the report.

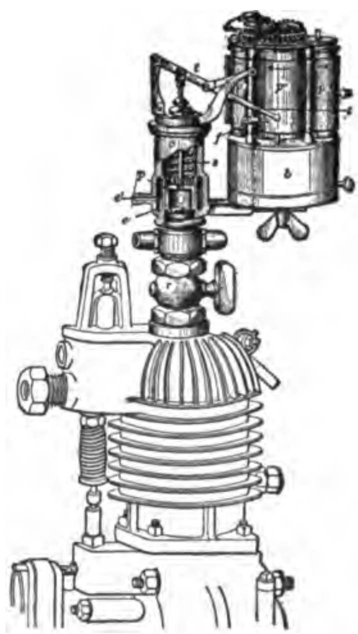


FIG. 16-4.

One method suggested for determining the instantaneous variation of speed which accompanies a change of load is as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of electricity. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the fly-wheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots.

When this is done the observer viewing the wheel through the slots sees what appears to be a stationary fly-wheel. When a change in the velocity of the fly-wheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the change of angular velocity during any given time is revealed.

Experiments that have been made with a device of this kind show that the instantaneous gain of velocity, upon suddenly removing all the load from an engine, amounted to from one-sixth to one-quarter of a revolution of the wheel.

In an engine which is governed by varying the number of explosions or working cycles, a record should be kept of the number of explosions per minute; or if the engine is running at nearly maximum load, by counting the number of times the governor causes a miss in the explosions.

One way of mechanically recording the explosions is to attach to the exhaust pipe a cylinder and piston arranged so that the pressure caused by the exhaust gases operate against a light spring and moves a register, which is provided for automatically counting the number.

AUTHOR'S NOTE. — An instrument for this purpose has been devised by R. Mathot. The following description is from his book on "Modern Gas Engines and Producer Gas Plants:"

The instrument, Fig. 16–4, is somewhat similar in form to the ordinary indicator. Its record, however, is made on a paper tape which is continuously unwound. The cylinder *c* is provided with a piston *p*, about the stem of which a spring *s* is coiled. A clock train contained in the chamber *b* unwinds the strip of paper from the roll *p'* and draws it over the drum *p''*, where the pencil *t* leaves the mark. The tape is then rewound on the spindle *p'''*. A small stylus or pencil *f* traces the atmospheric line on the paper as it passes over the drum *p''*. In order to obviate the binding of the piston *p* when subjected to the high temperature of the explosions, the cylinder *c* is provided with a casing *e* in which water is circulated by means of a small rubber tube which fits over the nipple *e'*. This recorder analyzes with absolute precision the work of all engines, whatever may be their speed. It gives a continuous graphic record from which the number of explosions, together with the initial pressure of each, can be determined, and the order of their succession. Consequently the regularity or irregularity of the variations can be observed and traced to the secondary influences producing them, such as the action of the inlet and outlet valves and the sensitiveness of the governor. It renders it possible to estimate the resistance to suction and the back pressure due to expelling the burnt gases, the chief causes of loss in efficiency in high-speed engines. Furthermore, the influence of compression is markedly shown from the diagram obtained.

The recorder is mounted on the engine; its piston is driven back by each of the explosions to a height corresponding with their force; and the stylus

or pencil controlled by the lever t records them side by side on the moving strip of paper. The speed with which this strip is unwound conforms with the number of revolutions of the engine to be tested, so that the records of the explosions are placed side by side clearly and legibly.

Their succession indicates not only the number of explosions and of revolutions which occur in a given time, but also their regularity, the number of misfires. The pressure of the explosions is measured by a scale connected with the recorder-spring. By employing a very weak spring which flexes at the bottom simply by the effect of the compression in the engine cylinder, it is possible to ascertain the amount of the resistance to suction and to the exhaust. It is simply sufficient to compare the explosion record with the atmospheric line, traced by the stylus f. By means of this apparatus, and of the records which it furnishes, it is possible analytically to regulate the work of an engine, to ascertain the proportion of air, gas, or hydrocarbon which produces the most powerful explosion, to regulate the compression, the speed, the time of ignition, the temperature, and the like.

XIV. Recording the Data. — The time of taking weights and every observation should be recorded, and note made of every event, however unimportant it may seem to be. The pressures, temperatures, meter readings, speeds, and other measurements should be observed every 20 or 30 minutes when the conditions are practically uniform, and at more frequent intervals if they are variable. Observations of the gas or oil measurements should be taken with special care at the expiration of each hour, so as to divide the test into hourly periods, and reveal the uniformity, or otherwise, of the conditions and results as the test goes forward.

All data and observations should be kept on suitable prepared blank sheets or in notebooks.

XV. Uniformity of Conditions. — When the object of the test is to determine the maximum economy, all the conditions relating to the operation of the engine should be maintained as constant as possible during the trial.

XVI. Indicator Diagrams and their Analysis. — SAMPLE DIAGRAMS: Sample diagrams nearest to the mean should be selected from those taken during the trial and appended to the tables of the results. If there are separate compression or feed cylinders, the indicator diagrams from these should be taken and the power deducted from that of the main cylinder.

XVII. Standards of Economy and Efficiency. — The hourly consumption of heat, determined as pointed out in Article IX, divided by the indicated or the brake horse-power, is the standard expression of engine economy recommended.

In making comparisons between the standard for internal

combustion engines and that for steam, it must be borne in mind that the former relates to energy concerned in the generation of the force employed, whereas in the steam engine it does not relate to the entire energy expended during the process of combustion in the steam boiler. The steam engine standard does not cover the losses due to combustion, while the internal combustion engine standard, in cases where a crude fuel such as oil is burned in the cylinder, does cover these losses. To make a direct comparison between the two classes of engines considered as complete plants for the production of power, the losses in generating the working agent must be taken into account in both cases and the comparison must be on the basis of the fuel used; and not only this, but on the basis of the same or equivalent fuel used in each case. In such a comparison, where producer gas is used, and the producer is included in the plant, the fuel consumption, which will be the weight of coal in both cases, may be directly compared.

The thermal efficiency ratio per indicated horse-power or per brake horse-power for internal combustion engines is obtained in the same manner as for steam engines, and is expressed by the fraction

$$\frac{2545}{\text{B.T.U. per H.P. per hour}}$$

XVIII. Heat Balance. — For purposes of scientific research, a heat balance should be drawn which shows the manner in which the total heat of combustion is expended in the various processes concerned in the working of the engine. It may be divided into three parts: first, the heat which is converted into the indicated or brake work; second, the heat rejected in the cooling water of the jackets; and third, the heat rejected in the exhaust gases, together with that lost through incomplete combustion and radiation.

To determine the first item, the number of foot-pounds of work performed by, say, one pound or one cubic foot of the fuel is determined; and this quantity divided by 778, which is the mechanical equivalent of one British thermal unit, gives the number of heat units desired. The second item is determined by measuring the amount of cooling water passed through the jackets, equivalent to one pound or one cubic foot of fuel consumed, and

calculating the amount of heat rejected, by multiplying this quantity by the difference in the sensible heat of the water leaving the jacket and that entering. The third item is obtained by the method of differences; that is, by subtracting the sum of the first two items from the total heat supplied. The third item can be subdivided by computing the heat rejected in the exhaust gases as a separate quantity. The data for this computation are found by analyzing the fuel and the exhaust gases, or by measuring the quantity of air admitted to the cylinder in addition to that of the gas or oil.

For methods of making fuel and exhaust gas computations, see Chapter VI.

XIX. Report of Test. — The data and results of a test should be reported in the manner outlined in one of the following tables, the first of which gives a complete summary when all the data are determined, and the second is a shorter form of report in which some of the minor items are omitted.

XX. Temperatures Computed at Various Points of the Indicator Diagram. — The computation of temperatures corresponding to various points in the indicator is, at best, approximate. It is possible only where the temperature of one point is known or assumed, or where the amount of air entering the cylinder along with the charges of gas or oil, and the temperature of the exhaust gases, is determined.

If the amount of air is determined for a gas engine, together with the necessary temperatures, so that the volume and the temperature of the air entering the cylinder per stroke, and that of the gas are known, we may, by combining this with the other data, compute the temperature for a point in the compression curve. In this computation we must allow for the volume of the exhaust gases remaining in the cylinder at the end of the stroke. The temperature at the point in the compression curve where it meets or crosses the atmospheric line will be given by the formula:

$$T = \frac{491.4\ V'}{V'' + V''' + V''''} - 459.4 \ldots . (A);$$

where V' is the total volume corresponding to the point where the compression curve meets or crosses the atmospheric line; V''

the volume of the air at atmospheric pressure entering the cylinder during each working cycle, reduced to the equivalent volume at 32 degrees Fahrenheit; V''' the volume of the gas consumed per cycle reduced to the equivalent at atmospheric pressure and 32 degrees Fahrenheit; and V'''' the volume of the exhaust gases retained in the cylinder reduced to the same basis. To reduce the actual volumes to those at 32 degrees Fahrenheit, multiply by the ratios of $491.4 \div (T' + 459.4)$, where T' is the observed temperature of the air and of the gas used as fuel. For the exhaust gases retained in the cylinder at the end of the stroke T' may be taken as the temperature of the exhaust gases leaving the engine, provided the engine is not of the "scavenging" type.

Having determined the temperature of a point in the compression curve, the temperature of any point in the diagram may be found by the equation

$$T_1 = (T + 459.4)\frac{P_1 V_1}{PV} - 459.4 \ldots. (B).$$

Here T_1 is the desired temperature of any point in the diagram where the absolute pressure is P_1 and the total volume V_1, and P and V are the corresponding quantities for the point in the compression line having the temperature T computed from the formula (A).

Formula (B) holds only where the weight of the gases contained in the cylinder is constant. It is also assumed in this formula that the density of the gas compared to air at the same temperature and pressure is the same before and after the explosion.

A second method may be employed, provided the air which enters the cylinder is measured. This will allow for any difference in the density of the gas before and after explosion, and more exact values for temperatures on the expansion curve may be obtained than by the first method.

In this method the density of the exhaust gases compared to air at the same temperature and pressure is computed, assuming perfect combustion, and including the effect of the water vapor present; and from this density the volume of the gases exhausted per cycle is determined. If this volume exhausted per cycle, added to the volume of the gas retained in the clearance space at the end of the stroke, be called V in equation B, and T be the

observed temperature of the exhaust gases, this equation may be used for determining the temperature of any point in the diagram in the way already described. This method is more complicated than the first, as it involves the determination of the theoretical density after explosion, but it possesses the advantage that it may be applied to an oil as well as to a gas engine.

A third method of computing the temperature of the various points in the diagram may be employed where analyses of the exhaust gases as well as of the fuel have to be made. This method is more complicated than the first, but, in common with the second, it possesses the advantage that it may be applied to an oil as well as to a gas engine.

In applying the third method the volume of the exhaust gases discharged per working cycle would be given by the formula:

$$V_2 = \frac{1}{D}(Rw + w) \ldots . (C)$$

where D is the density of the exhaust gases at their observed temperature, computed from the analysis, assuming the vapor of water produced through burning the hydrogen in the fuel to be in a gaseous state; R the weight of the air which enters the cylinder per pound of fuel consumed per working cycle; the value of R, providing there are no unconsumed hydrocarbons, may be computed by employing the formula:

$$R = \frac{\mathrm{NC}}{.33\,(\mathrm{CO_2} + \mathrm{CO})} \ldots . (D)$$

where N, CO_2 and CO represent the proportions, by volume, of the several constituents of the exhaust gases, and C the weight of carbon consumed and converted to CO_2 or CO per pound of fuel burned, computed from the analysis of the fuel and of the exhaust gases.

Having determined the volume V_2 of the exhaust gases, formula (B) may be used in computing the temperature, in which case T will represent the temperature of the exhaust gases as in the second method, P the pressure of the exhaust, and V the volume of the exhaust gases V_2 discharged per stroke, added to the volume of the gases retained in the cylinder at the end of the stroke.

The value of R given in equation (D) is approximate, on

account of the fact that the percentage of N should be that due to the air alone, and not that due to the air in addition to that contained in the fuel gas. Where extreme accuracy is desired, the value found for R may be used to determine the percentage of N which in the analysis of the exhaust gases is due to the N in the fuel gas, and this value may be subtracted from the total N shown by the analysis of the fuel gases, in order to obtain the correct value of N to be used in equation (*C*).

TABLE NO. 1

DATA AND RESULTS OF TEST OF GAS OR OIL ENGINE

Arranged according to the Complete Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902

1. Made by … of …
   on engine located at …
   to determine …
   …
2. Date of trial …
3. Type of engine, whether oil or gas …
   …
4. Class of engine (mill, marine, motor for vehicle, pumping, or other) …
   …
5. Number of revolutions for one cycle, and class of cycle …
   …
6. Method of ignition …
   …
7. Name of builders …
8. Gas or oil used …
   (*a*) Specific gravity … deg. Fahr.
   (*b*) Burning point … "
   (*c*) Flashing point … "
9. Dimensions of engine:

		1st Cyl.	2d Cyl.
(*a*) Class of cylinder (working or for compressing the charge) …			
(*b*) Vertical or horizontal …			
(*c*) Single or double acting …			
(*d*) Cylinder dimensions …			
Bore …	in.		
Stroke …	ft.		
Diameter piston rod …	in.		
Diameter tail rod …	in.		
(*e*) Compression space or clearance in per cent of volume displaced by piston per stroke …			
Head end …			
Crank end …			
Average …			
(*f*) Surface in square feet (average) …			
Barrel of cylinders …			
Cylinder heads …			
Clearance and ports …			
Ends of piston …			
Piston rod …			

(g) Jacket surfaces or internal surfaces of cylinder heated by jackets, in square feet ....
Barrel of cylinder........................
Cylinder heads ....
Clearance and ports ....
(h) Horse-power constant for one lb. M. E. P., and one revolution per minute ....

10. Give description of main features of engine and plant, and illustrate with drawings of same given on an appended sheet. Describe method of governing. State whether the conditions were constant throughout the test.

Total Quantities

11. Duration of test .... hours.
12. Gas or oil consumed .... cu. ft. or lbs.
13. Air supplied in cubic feet .... cubic feet.
14. Cooling water supplied to jackets .... "
15. Calorific value of gas or oil by calorimeter test, determined by ....calorimeter .... B. T. U.

Hourly Quantities

16. Gas or oil consumed per hour .... cu. ft. or lbs.
17. Cooling water supplied per hour........................ lbs.

Pressures and Temperatures

18. Pressure at meter (for gas engine) in inches of water...... ins.
19. Barometric pressure of atmosphere:
(a) Reading of height of barometer .... "
(b) Reading of temperature of barometer .... deg. Fahr.
(c) Reading of barometer corrected to 32° Fahr. .... ins.
20. Temperature of cooling water:
(a) Inlet .... deg. Fahr.
(b) Outlet .... "
21. Temperature of gas at meter (for gas engine) .... "
22. Temperature of atmosphere:
(a) Dry-bulb thermometer .... "
(b) Wet-bulb thermometer .... "
(c) Degree of humidity .... per cent.
23. Temperature of exhaust gases .... deg. Fahr.
How determined ....

Data Relating to Heat Measurement

24. Heat units consumed per hour (lbs. of oil or cu. ft. of gas per hour multiplied by the total heat of combustion)..... B. T. U.
25. Heat rejected in cooling water:
(a) Total per hour .... "
(b) In per cent of heat of combustion of the gas or oil consumed .... per cent.
26. Sensible heat rejected in exhaust gases above temperature of inlet air:
(a) Total per hour .... B. T. U.
(b) In per cent of heat of combustion of the gas or oil consumed .... per cent.
27. Heat lost through incomplete combustion and radiation per hour:
(a) Total per hour .... B. T. U.
(b) In per cent of heat of combustion of the gas or oil consumed .... per cent.

Speed, Etc.

28. Revolutions per minute .... rev.
29. Average number of explosions per minute ....
    How determined ....
30. Variation of speed between no load and full load .... rev.
31. Fluctuation of speed on changing from no load to full load measured by the increase in the revolutions due to the change.

Indicator Diagrams

	1st Cyl.	2d Cyl.
32. Pressure in lbs. per sq. in. above atmosphere:		
(a) Maximum pressure		
(b) Pressure just before ignition		
(c) Pressure at end of expansion		
(d) Exhaust pressure		
33. Temperatures in deg. Fahr. computed from diagrams:		
(a) Maximum temperature (not necessarily at maximum pressure)		
(b) Just before ignition		
(c) At end of expansion		
(d) During exhaust		
34. Mean effective pressure in lbs. per sq. in		

Power

35. Power as rated by builders:
    (a) Indicated horse-power........................... H. P.
    (b) Brake .... "
36. Indicated horse-power actually developed:
    First cylinder .... "
    Second cylinder .... "
    Total .... "
37. Brake H. P., electric H. P., or pump H. P., according to the class of engine .... "
38. Friction indicated H. P. from diagram, with no load on engine and computed for average speed .... "
39. Percentage of indicated H. P. lost in friction .... per cent.

Standard Efficiency Results

40. Heat units consumed by the engine per hour:
    (a) Per indicated horse-power .... B. T. U.
    (b) Per brake horse-power .... "
41. Heat units consumed by the engine per minute:
    (a) Per indicated horse-power .... "
    (b) Per brake horse-power .... "
42. Thermal efficiency ratio:
    (a) Per indicated horse-power .... per cent.
    (b) Per brake horse-power .... "

Miscellaneous Efficiency Results

43. Cubic feet of gas or lbs. of oil consumed per H. P. per hour:
    (a) Per indicated horse-power ....
    (b) Per brake horse-power ....

Heat Balance

44. Quantities given in per cents of the total heat of combustion of the fuel:

(*a*) Heat equivalent of indicated horse-power	....	per cent.
(*b*) Heat rejected in cooling water	....	"
(*c*) Heat rejected in exhaust gases and lost through radiation and incomplete combustion	....	"
	Sum = 100	"
Subdivisions of Item (*c*):		
(*c*1) Heat rejected in exhaust gases	....	"
(*c*2) Lost through incomplete combustion	....	"
(*c*3) Lost through radiation, and unaccounted for	....	"
	Sum = Item (*c*)......	

Additional Data

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean and the corresponding scales. Where analyses are made of the gas or oil used as fuel, or of the exhaust gases, the results may be given in a separate table.

TABLE NO. 2

DATA AND RESULTS OF STANDARD HEAT TEST OF GAS OR OIL ENGINE

Arranged according to the Short Form advised by the Engine Test Committee, American Society of Mechanical Engineers. Code of 1902.

1. Made by .... of ....
   on engine located at ....
   to determine ....
   ....
2. Date of trial ....
3. Type and class of engine ....
   ....
4. Kind of fuel used ....
   (*a*) Specific gravity .... deg. Fahr.
   (*b*) Burning point .... "
   (*c*) Flashing point .... "
5. Dimensions of engine:

	1st Cyl.	2d Cyl.
(*a*) Class of cylinder (working or for compressing the charge)		
(*b*) Single or double acting		
(*c*) Cylinder dimensions:		
Bore .... in.		
Stroke .... ft.		
Diameter piston rod .... in.		
(*d*) Average compression space, or clearance in per cent		
(*e*) Horse-power constant for one lb. M. E. P. and one revolution per minute		

Total Quantities

6. Duration of test .... hours.
7. Gas or oil consumed .... cu. ft. or lbs.
8. Cooling water supplied to jackets ....
9. Calorific value of fuel by calorimeter test, determined by .... calorimeter .... B. T. U.

Pressures and Temperatures

10. Pressure at meter (for gas engine) in inches of water......	ins.
11. Barometric pressure of atmosphere:	
(a) Reading of barometer	"
(b) Reading corrected to 32 degs. Fahr	"
12. Temperature of cooling water:	
(a) Inlet	deg. Fahr.
(b) Outlet	"
(c) Degree of humidity	"
13. Temperature of gas at meter (for gas engine)	"
14. Temperature of atmosphere:	
(a) Dry bulb thermometer	"
(b) Wet bulb thermometer	"
15. Temperature of exhaust gases	"

Data Relating to Heat Measurement

16. Heat units consumed per hour (pounds of oil or cubic feet of gas per hour multiplied by the total heat of combustion)	B. T. U.
17. Heat rejected in cooling water per hour	"

Speed, Etc.

18. Revolutions per minute	rev.
19. Average number of explosions per minute	

Indicator Diagrams

20. Pressure in lbs. per sq. in. above atmosphere:

	1st Cyl.	2d Cyl.
(a) Maximum pressure		
(b) Pressure just before ignition		
(c) Pressure at end of expansion		
(d) Exhaust pressure		
(e) Mean effective pressure		

Power

21. Indicated horse-power:	
First cylinder	H. P.
Second cylinder	"
Total	"
22. Brake horse-power	"
23. Friction horse-power by friction diagrams	"
24. Percentage of indicated horse-power lost in friction	per cent.

Standard Efficiency, and Other Results

25. Heat units consumed by the engine per hour:	
(a) Per indicated horse-power	B. T. U.
(b) Per brake horse-power	"
26. Pounds of oil or cubic feet of gas consumed per hour:	
(a) Per indicated horse-power	lbs. or cu. ft.
(b) Per brake horse-power	"

Additional Data

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is to be used. Also give copies of indicator diagrams nearest the mean, and the corresponding scales.

RULES FOR TESTING GAS PRODUCERS AND GAS ENGINES. CODE OF THE GERMAN SOCIETY OF ENGINEERS *

All metric units have been transposed to English units

The preparation of the following rules for making gas engine and producer tests was undertaken by a committee appointed from the Verein deutscher Ingenieure, in collaboration with the German Society of Engine Builders, with the view of establishing definite general regulations governing such tests. It is desirable, by specifying the important proportions of the examined plants and the conditions under which the results were attained, to insure that these results are not only applicable to a single case, but that they have general value. To attain this end it is necessary that all data should be given uniformly according to a code of regulations such as that here presented.

The execution of such tests should be intrusted only to persons possessing the required expert knowledge and practical experience. These persons must make a trial plan, or schedule, appropriate to the individual case in hand, which, in many instances, will not require that all of the investigations stipulated in the general code are actually carried out. They must further examine the instruments for measuring or recording purposes as to their fitness and must compile the results. The following rules, the adoption or selection of which must be left to the soundness of judgment of the investigator, are intended to serve as a basis on which to proceed.

GENERAL REGULATIONS

Object of Investigation

1. The object of a test made on a producer-gas plant may be to determine:

(*a*) The quantity, composition, and calorific value of the fuel consumed.

(*b*) The quantity, composition, and heat value of the gas produced.

(*c*) The degree of efficiency of the producer-gas plant.

(*d*) The separate heat losses in the plant.

* Mainly from F. E. Junge's translation in Power, Feb., 1907.

(*e*) The quantity of impurities contained in one cubic meter or one cubic foot of gas (dust, tar, sulphur, etc.).

(*f*) The moisture contents of the gas.

(*g*) The water consumption of the producer-gas plant, either total or in the separate parts.

(*h*) The mechanical work required for operating the plant, including apparatus.

(*i*) The duration of time required for starting.

(*k*) The stand-by losses during intervals of shutting down day or night.

2. The object of a test made on an internal combustion (gas) engine may be to determine:

(*a*) The indicated capacity and the effective output.

(*b*) The mechanical efficiency.

(*c*) The fuel consumption and the heat consumption per horse-power hour.

(*d*) The consumption of lubricants, separately for cylinder and engine.

(*e*) The consumption of water and the heat conducted to the cooling water.

(*f*) The fluctuations in number of revolutions.

(*g*) The composition of exhaust gases.

NUMBER AND DURATION OF TESTS

Admissible Fluctuations

3. The number and duration of trials are determined by the purpose of the test as well as by a consideration of the conditions of installation and operation, and must be settled and previously arranged according to paragraphs four to eight. For trials of special importance the results of which are decisive for acceptance tests, for penalties or for premiums, this item deserves special consideration.

4. Acceptance tests should be made if possible immediately after a plant has been put into actual operation; the manufacturers, however, must be granted a reasonable time for making preliminary trials of their own and for carrying out alterations or improvements then necessary. The length of this time and other conditions are best agreed upon when drawing up the delivery contract.

5. In order to be able to get acquainted with the operation of the plant that is to be tested, to find time for examining the testing devices employed, and to break in observers and assistants, it is desirable that preliminary trials be allowed.

6. If the fuel consumption in gas producers is to be determined, the trial run must be extended over at least eight hours under constant conditions and without interruptions.

7. For determining the consumption of liquid or gaseous fuel and provided the conditions are constant, it is sufficient for the higher loads to extend measurements over an hour, while for finding the consumption at the lower loads, measurements of even shorter duration are sufficient. To ascertain the constancy of the conditions the temperature of the outflowing cooling water must be read from time to time. These rules as to the duration of the tests are made with the provision that no interruption or disturbance of the trial takes place, and that intermediate readings show only slightly varying values for the consumption.

8. If only the mechanical efficiency of an engine is to be determined, trials of short duration under constant conditions are sufficient; but at least ten sets of indicator cards should be taken.

9. For investigations of special importance at least two tests should be made, one after the other. They should be accepted only if no interruptions occurred and if the results show no greater deviations than those due to unavoidable errors of observation. The mean of the two results is to be taken as the final result.

10. The extent to which the capacity and consumption of gas may differ from the guarantee or contract figures, without justifying a claim of breach of contract, is to be clearly stated before the tests (either in the original contract or in the schedule of tests). When no other agreement has been previously arrived at, the capacity guarantee is regarded as fulfilled if the figure obtained in the test is not more than 5 per cent below the value on which the guarantee was based. This margin, however, is allowable only for the maximum output which was promised beyond the guaranteed continuous output. The latter must be rendered by the engine under all circumstances.

The consumption of fuel and water as determined on test should not exceed the guaranteed figures by more than 5 per cent

even if, during the trial, the engine load fluctuated somewhat from the load upon which the guarantee was based, provided that fluctuation do not exceed an average of ±5 per cent of such load, or a maximum of ±15 per cent.

Since it is often impossible when making tests to have the internal combustion engine work at exactly the effective (horsepower) capacity on which the guarantee agreed upon in the contract is based, it is recommended that the agreement shall specify the expected fuel consumption for higher and lower outputs. The same provision is perferably made also with gas producers.

UNITS OF MEASUREMENT AND DESIGNATIONS

11. When giving pressure data it must be stated whether absolute pressures or gage pressures above or below the atmospheric are meant. Absolute pressure equals atmospheric pressure plus gage pressure.

12. All temperature and heat measurements refer to the Fahrenheit scale.

13. The mechanical equivalent of heat is taken at 778 foot-pounds.

14. The calorific value of a fuel is to be taken as its lower heating value; that is, the heat which is liberated by the complete combustion of the fuel when the burnt products are cooled down to the original (room) temperature at constant pressure, it being assumed, however, that the water of combustion and the moisture contained in the fuel remain vaporized. The calorific value must be based on the unit quantity or weight of original fuel, without deducting ash, moisture, etc., and is to be expressed in heat units. For both solid and liquid fuels the unit of weight is the pound.

The heat value of gaseous fuels is based on one cubic foot at 32 degrees Fahrenheit, and 760 millimeters barometer pressure, or must be expressed in thermal units as "effective" heat value, that is, reduced to one cubic foot of actual gas used. If not specially stated, it is always understood that the heat value recorded is that of gas at 32 degrees Fahrenheit and 760 millimeters barometer pressure.

In this country the general standard so far recommended seems to indicate for "standard gas" a temperature of 60 degrees

Fahrenheit, and a pressure of 14.7 pounds per square inch, corresponding to the usual atmospheric pressure.

15. The efficiency of a gas-producer plant is the ratio of the latent heat contained in the gas as produced to the heat of combustion of the total weight of fuel consumed in the plant, both items being computed from the lower heating value. In producer-gas plants having a separately fired steam boiler, it is advisable also to determine the ratio of the heat which is chemically bound in the producer gas to the heat equivalent of that portion of the fuel which is consumed in the producer proper for making such gas.

16. The unit of measurement used for the power or work output of an internal combustion engine is the horsepower equal to 33000 foot-pounds per minute. It must be clearly stated whether the indicated power, or the useful or available power, is meant. If not otherwise designated it is understood that the figures refer to the useful or available output.

17. The indicated power of the engine, or the indicated work, is the difference between the total power developed or work done, and the indicated power, or work, which is consumed within the engine; in short, the difference between the positive and the negative indicated power or work.

AUTHOR'S NOTE.—This is the provision which caused considerable discussion among gas-engine experts some time ago. It means as it stands, that in a 4-cycle machine, the indicated horse-power is that determined from the work diagram minus the work shown by the lower loop diagram; and, in a 2-cycle engine, the total indicated horse-power as determined from the diagram of the power cylinder minus the pump work is considered as the indicated horse-power. This view is undoubtedly correct when the mechanical efficiency of the engine itself as a machine is to be determined.

The power required at "no load" is the power indicated when no useful work is rendered by the engine.

18. Mechanical efficiency is the ratio of the useful power to the indicated power of the engine.

19. All consumption figures should be reduced to the hour basis, and if they are to be compared with the output of the engine they must be based on one horse-power hour. If not otherwise agreed upon, these data refer to the useful or available output at full load.

EXECUTION OF TESTS

20. If the quantity of gas made in a producer or the weight of fuel consumed in an engine is to be measured, then all pipes or ducts which are not used in the test must be cut off from the piping which leads to the producer and engine that are to be tested. This is best done by means of blind flanges. The active ducts, pipes, gas holders, etc., must be examined with regard to leakage and made tight if necessary. Unavoidable losses due to leakage must be determined. This holds especially for masonry gas mains.

FUEL CONSUMPTION OF A GAS-PRODUCER PLANT

21. The kind, number, and duration of tests must be agreed upon according to the general rules laid down in paragraphs 1 to 10.

22. The constructive features and the operative conditions of gas-producer plants must be described and illustrated in the report by drawings, so far as this is necessary, to arrive at a clear understanding of the manner of working and of the results obtained.

23. Before making the test the plant should be examined as to whether or not it is in good working order.

24. The quantity of fuel consumed in the gas producer is determined by taking the weight of the fuel which is charged into the producer during the trials in order that the producer may contain at the end of the test exactly the same amount of heat — either liberated, or chemically bound in the fuel — that it contained when starting the test. To meet this requirement it is not sufficient that the depth of the fuel bed be the same at the end as it was at the beginning; it must also be taken into consideration what influence the ash and the slag left in the producer, the location of the incandescent zone, the formation of fissures and cavities, the closeness or density of the producer charge, and the chemical composition of the burning fuel particles exercise on the heat contents of the producer.

In order to comply with this requirement the following rules should be followed:

25. When starting the test the plant should be in the condition of stability or normal working condition, if possible.

This means that after a period of shut-down for cleaning or repairs it should be in active operation for one or more days, running on fuel of the same characteristics and size, with the same depth of fuel bed, the same skill of attendance as regards the charging or feeding of fresh fuel and the removing of slag, and under the same load conditions that will obtain during the test.

26. During the trial the producer should be charged and poked as nearly in accordance with the requirements for attendance as possible. The level of fuel charged must be the same at the beginning and at the end of the tests and should be kept constant during the trial. About half an hour before starting and before stopping a test, the slag and ashes should be removed.

If it is impossible to rake out the ashes during the operation of the producer, the plant must be shut down immediately after stopping the test, the ashes must be taken out at once and the producer refilled up to the same level that existed when starting the test. The weight of fuel used for this purpose must be added to the consumption.

27. The fuel consumed during the trial must be weighed, also the fuel which has not been burnt and remains useful; that is, that portion which drops down from above the grate while raking out the ashes, and that which is culled out from the ashes as unburnt. The weight of the former may be deducted from the consumption, but not the amount which is taken out from the ashes, nor the coal dust which accumulates in the scrubbers and in the flues between the producer and the engine.

28. To be able to determine the quantity of ash and slag produced during the trial, the ash box must be emptied before the test. If this is not possible, as when an inclined grate is used, the refuse in the ash box must be equalized before and after the run.

29. The stand-by losses during intervals of shutting down at day and night must be determined.

30. In order to get a representative sample of the solid fuel, the following course may be pursued: Of every carload, basket, or other measure of fuel, put a shovelful into a covered receptacle. Immediately after the test is over, the contents of the receptacle should be broken, mixed, spread and quartered by drawing the

two diagonals of a square. The two opposite quarters are rejected, the two others broken up finer, mixed and quartered, and the two opposite quarters rejected. This is continued until a sample of some 10 to 20 pounds remains, which is preserved, in well-closed receptacles, for analysis. In addition to this a number of other samples must be put away in air-tight receptacles for use in determining the contents of moisture in the fuel.

31. The composition of the fuel shall be determined by chemical analysis. Its contents in carbon (C), hydrogen (H), oxygen (O), sulphur (S), ash (A), and water (W) must be given in percentage of weight referred to the original fuel. The contents, in the fuel, of nitrogen (N) can be disregarded. The behavior of the fuel when being heated should be determined by a coking test.

32. The calorific value of the fuel must be determined by calorimetric analysis. An approximate determination of the heating value can be made on the basis of the chemical analysis by employing De Long's formula:

$$\text{Heating value} = 145\ C + 522.3\left(H - \frac{O}{8}\right) + 40\ S - 9.66\ W$$

in which C, O, H, S, and W are expressed in weight per cent.

TESTING AN INTERNAL-COMBUSTION ENGINE

33. Kind, number, and duration of trials to be agreed upon according to the general regulations Nos. 1 to 8.

34. The constructive features and operative conditions of the engine must be so illustrated in the report as to enable one to form a correct idea of the manner of working and of the results of operation. Especially important are the type and capacity of engine, diameter of cylinder and piston-rod, piston stroke, contents of clearance space, and other essential dimensions; the normal rate of revolution and the admissible fluctuations; kind and heat value of fuel for which the engine is intended. The diameter of the cylinder and the stroke should be actually measured if this is possible.

The contents of the compression space are preferably determined by filling with water. If it is impossible to state the cubical contents of the compression space, then the compression

pressure at full load should at least be given. This is done by taking an indicator card while the ignition is interrupted.

35. Before making the test the engine must be examined internally and externally as to whether or not it is in good working order.

36. The number of revolutions of the engine should be determined by a continuous speed counter, the records of which must be noted at certain intervals, and must be checked or corrected from time to time by direct readings. If the speed conditions of the engine are to be investigated it is essential to determine the following items:

(a) The number of revolutions under constant conditions at maximum load and at no load.

(b) The fluctuations in speed at constant load.

(c) The temporary change in the number of turns when the load is suddenly decreased or increased from a given constant load by a prescribed amount. These determinations can be executed with apparatus of the character of the Horn tachograph. The fluctuations of speed during the performance of one engine cycle above and below the mean value, expressed in parts of the latter, should be determined by calculation unless otherwise provided.

The coefficient of fly-wheel regulation is

$$\delta = \frac{N_{\text{max.}} - N_{\text{min.}}}{\frac{N_{\text{max.}} + N_{\text{min.}}}{2}} = 2\left(\frac{N_{\text{max.}} - N_{\text{min.}}}{N_{\text{max.}} + N_{\text{min.}}}\right)$$

where N = number of revolutions.

37. The useful output can be determined either by brake test or by electrical measurement.

The dimensions and weight of the brake should be determined before the trial.

The electrical measurements can be made on a generator directly coupled to the gas engine. The useful work is computed from the output of the dynamo. The efficiency of the generator should be determined by one of the methods as laid down in the "Rules for Judging and Testing Electrical Machinery and Transformers," published by the association of German electrical engineers. If the efficiency is found approximately by measuring

the determinable losses, then an adequate amount (say 2 per cent of the full load output) must be allowed for losses not accounted for.

The apparatus with which the electrical measurements are executed must be calibrated before and if possible also after the test.

Whether anything besides the 2 per cent above allowed should be credited to the gas engine for increased bearing friction and windage of the generator must be settled in each individual case.

Whether, in case the useful output can neither be determined by brake test or by electrical measurements, the code provision for testing steam engines can be admitted as correct for gas engines, namely, to designate the useful output as the difference between the indicated work at any load and the indicated work at no load, cannot be settled at the present state of development, since results of accurate investigations are not yet available.

38. Indicators must be connected immediately to the combustion chamber without employing long piping with sharp bends, and one indicator must be provided for every combustion chamber. For this purpose each compression chamber must have an opening for three-quarter or one inch Whitworth thread. The same holds true for pump cylinders.

The indicators and their springs must be calibrated before and after the test according to the accepted standards.

39. During the test, cards should be taken quite frequently from every combustion chamber and from the pump cylinders. The cards should be designated by numbers, and the time when each card was taken, the scale of springs used and the number of single cards obtained must be recorded on the cards. At least five diagrams should be taken on one card successively. From time to time diagrams indicated with a weak spring should be taken from the combustion chambers.

The indicated work at no load should be determined immediately after stopping the main test and while the engine is still warmed up ready for operation. Care must be taken that the no-load cards are not taken during an acceleration or during a retardation period of the fly-wheel.

ANALYSIS OF THE GAS GENERATED IN A PRODUCER-GAS PLANT OR CONSUMED IN AN INTERNAL COMBUSTION ENGINE, OR OF THE LIQUID FUEL USED

40. The samples for the chemical analysis of the gas must be taken during the trial at regular intervals and as frequently as possible.

They must be either analyzed on the spot or preserved in glass tubes closed by melting the ends. The analysis is to determine, in per cent of volume, the contents of the gas in carbon monoxide (CO), carbon dioxide (CO_2), hydrogen (H_2), marsh gas (CH_4), heavy hydrocarbons and oxygen (O_2).

In addition it is recommended to determine the contents of sulfur. The gas samples should be taken from the gas main between the cleaning apparatus and the engine.

41. The heat value of the gas should be determined as often as possible by calorimetric analysis, and the burner of the calorimeter should be fed from the gas main without interruption. In suction producer plants this can be done by means of a gas pump drawing from the main. If conditions should make it necessary that a sample be taken from the pipe while the calorimeter is shut off, such sample to be later transferred to and burned in the calorimeter, then the quantity of gas so taken should not be less than 300 liters (10.59 cubic feet), in order that the calorimeter may at first be brought into the condition of stability as regards the water of combustion, and in order that at least 100 liters (3.53 cubic feet) remain available for two successive analyses. The suction pump, the gas holder and the piping must be made tight with special care when making a calorimeter analysis of suction gas.

42. The gas meter of the calorimeter in which the heat value of the gas is determined must be calibrated. For determining the temperatures of the calorimeter water, only thermometers with calibration certificates or others compared with such should be used. The scales must be divided at least into tenths of a degree.

On the basis of the chemical analysis the heating value per standard cubic foot of gases which do not contain heavy hydro-

carbons can be computed from the following formula, if a calorimetric analysis cannot be made

$$\text{Heating Value} = 3.42\ CO + 2.97\ H_2 + 9.52\ CH_4$$

where CO, H_2 and CH_4 are expressed in volume per cent.

43. The quantity of gas produced or consumed should be measured by means of a gas holder or a gas meter. The cross-sectional area of the holder should be determined by measurement of its circumference at several places. Consumption tests with the gas holder shall not be made while the latter is exposed to the sun.

44. The gas meter must be calibrated and set level; it must be so filled that the water level corresponds to the normal filling existing during calibration. Between the gas meter and the engine a pressure regulator must be installed or a large suction space provided so that the water level shows only small pulsations during the pressure fluctuations.

45. At intervals corresponding to the duration of test the following readings should be taken: Position of the bell of the gas holder at three places or the records shown by the gas meter; the pressure in the bell or in the gas meter; the temperature of the gas when entering and when leaving the gas holder or the gas meter and before reaching the engine; the barometric pressure.

46. If the temperature of the gas is different when measuring the consumption than when measuring the heat value, the computation must also take into account the increase of volume which is due to the moisture contents of the gas at higher temperatures.

47. The consumption of liquid fuel must be determined either by weight or by measuring its volume. For determining heat value, composition, and specific weight of the fuel one representative average sample is sufficient.

48. When measuring the fuel consumption of internal combustion engines, the consumption of lubricating oil for the cylinder should be determined at the same time.

49. If the consumption at low loads of a double-acting tandem or twin engine is to be determined, it is not allowable to shut off the gas from one or more ends of the cylinders, provided that no other arrangements have been previously agreed upon and

are mentioned in the report, or that the governor acts automatically in the way described.

EXPLANATIONS TO VARIOUS ARTICLES OF THE CODE

The main code is followed by a number of explanations from which the following extracts are taken. The figures refer to the paragraphs of the above code.

1 & 2

In most cases only one or two of the objects of test mentioned are taken into account in any given trial. If in any exceptional case the object of the test should not be any of those mentioned, it should be a simple matter to adapt the rules given.

Under 2 (*c*) the term horse-power hour is used. It is essential that in any given trial this term be more closely defined, as horse-power may mean indicated brake, or even horse-power developed by pumps.

4

It is extremely desirable that the contract state the time allowed the manufacturer for adjustment and trial runs, because his own interests may make him call sometimes for a long, sometimes for a short period. In the case of a small engine, more or less a commercial stock machine, he may wish to have the period as short as possible, and this the buyer may agree to without danger of loss to himself. If, however, the machine is of a special type, or one provided with special attachments, it is but a matter of justice to allow the manufacturer a reasonable time in which to break in the engine and to give him an opportunity to correct any imperfections that may appear. It is to the interest of the buyer to grant such a period in order to become familiar with the machine before taking over the entire responsibility of operating it. It is also true that many faults appear only after some weeks of operation.

On the other hand, too long a period of adjustment is in many cases not acceptable to the buyer, because any extended work of improvement usually seriously hampers operation; and because in many cases he desires an operative machine, which no longer requires the care of the manufacturer, as soon as possible.

It frequently happens that no acceptance test is agreed upon.

In such cases it sometimes happens that the buyer comes back upon the manufacturer for faults which did not develop until the machine had been in operation some time.

If the manufacturer then agrees to an investigation or a test, a sufficient period should be given him to make any investigation he sees fit or to correct any imperfections that may have appeared before the decisive trial or investigation is made. This sometimes leads to a simple settlement of the matter in that the manufacturer discovers that ignorance or carelessness on the part of the operator have caused the imperfections complained of. The granting of such a period also guards the buyer against any later claim of the manufacturer that during the trial the machine was not in the condition in which he delivered it.

5

Preliminary tests are always desirable, but not absolutely necessary. The cost of any kind of investigation is usually quite high and of course the cost increases directly with the time. The expert called will therefore make such tests when they seem to him essential. But the manufacturer should have the right to call for the time necessary for such trials if he is to present the machine in its best condition.

6

It cannot be denied that eight hours is a rather short time, because it is extremely difficult to determine whether the producer is in the same condition at the end as at the beginning of the test, and because this uncertainty may lead to large errors. On the other hand it is unquestionable that in many cases a longer time would call forth so many difficulties in operation that eight hours would seem the necessary limit.

The rule is mainly framed to prevent trials of so short duration that serious errors can hardly be avoided, but it leaves it to the judgment of the experimenter whether to make the tests longer than eight hours where it seems desirable and is possible to do so.

7

Intermediate readings are recommended without qualification, since they form the best criterion of the constancy of conditions.

With liquid or gas fuels of constant composition, individual readings every five minutes apart sometimes show no variation for hours at a time. In such a case it is useless to extend the time of the trial.

8

In determining the mechanical efficiency of an engine it should not be forgotten that, although the average load may be constant, there may be speed variations due to the inevitable inequality of the indicator diagrams, so that during some cycles work is done in accelerating the fly-wheel, while during others the fly-wheel by retardation gives up some of its kinetic energy.

To minimize any error that this may introduce into the determination of the mechanical efficiency, at least ten diagrams should be taken.

If the conditions are otherwise constant, however, it is not necessary to spread these diagrams over any considerable period of time.

It is self-evident that during the time of taking the diagrams the supply of lubricating oil must not be increased.

Changes in the mechanical efficiency of the engine, as for instance those due to fouling, cannot be detected with certainty even by a long test period; they become noticeable usually only after a period of operation extending over two weeks. The determination of the mechanical efficiency of an engine, after constant conditions of operation are attained, therefore only applies to the engine in its then existing state or condition.

The number of diagrams to be taken on one card cannot be definitely stated. On account of variation in the diagrams, which is less at high than at low loads, care should be had not to take too few. On the other hand it is useless to take more than can be clearly distinguished. The running together of a larger number of diagrams only makes their evaluation more uncertain.

10

In consideration of unavoidable errors of observation, possible errors of the instruments used, etc., it is meet and usual to allow a certain margin between the figures found on trial and those guaranteed. In the steam engine code 5 per cent is allowed for this, and it seems reasonable to assume the same figure in this

case. Only in one point, in the guaranteed normal capacity, does the gas engine call for an exception.

A given steam engine gives its most economical results at a certain cut-off, but a higher capacity can always be obtained at the expense of a little economy, that is, a buyer is certain that even a machine slightly too small will give him sufficient capacity. A gas engine, on the contrary, works with the greatest economy at its maximum load. It is to the interest of the buyer, therefore, to get an engine exactly suited to his needs and not to choose it too large. It is possible for the same reason that any engine, if lacking slightly in guaranteed capacity, may become absolutely useless to the buyer. For these reasons it was thought advisable not to grant the manufacturer any leeway whatever as regards guaranteed capacity.

It is clear, therefore, that the manufacturer must take upon himself any possible inaccuracies in the measurements, unless he can show them up and demand a new trial. For that reason it is well for him to make his guarantee a little on the safe side of what he knows his engine is capable of developing. On the other hand, there is no harm done to the interest of the buyer if the manufacturer underrates the normal capacity of his machine, because the former will always call for an engine of a certain normal capacity to suit his needs. If he fails to do this, but places his dependence in the guaranteed maximum capacity, he is open to the charge of carelessness.

Since during acceptance tests it is often not possible to keep the load quite constant, it became necessary, following the steam engine code, to allow a certain amount of variation, within which no just cause could be found for objection to the trial. There are cases where the variations occurring are much greater, as when a gas engine is used for driving a roll train. But no one set of "Rules" can possibly take into account all such extreme cases, and in such instances the contract should contain the necessary agreements to make any test clear and free from subsequent objections.

The wish has been expressed from several quarters, that the "Rules" should contain a definition of the term "Normal Capacity." On account of the peculiarity of the gas engine above discussed this is not quite feasible. But the term "Maximum

Continuous Capacity" perhaps defines most nearly what is intended in most cases.

14

It is sometimes the case that the heating value of the standard cubic foot, that is, reduced to 32 degrees Fahrenheit and 760 mm. barometer, is so greatly different from the actual value of the gas as used, that any contract which contains only the heating value of the gas stated on that basis does not convey much meaning to the non-technical buyer. If for instance a given gas has a heating value of 135 B. T. U. per standard cubic foot, its effective heating value at a high altitude and in a warm climate, say at 68 degrees and 620 mm. barometer, will only be about 100 B. T. U. per cubic foot. To obviate any misunderstanding, it should be clearly stated that, when the "effective" heating value of the gas is not definitely specified, the heating value at 32 degrees Fahrenheit and 760 mm. barometer is meant.

19

By "full" load is meant the normal capacity, as per paragraph 10.

23

For acceptance tests, and all other tests which are intended to decide any disagreements between manufacturer and buyer, such examination should be carried out in the presence and with the aid of the former, as already mentioned under paragraph 4,

24–26

In all gas producer tests it is hardly possible with certainty to have all conditions exactly the same at the end as at the beginning. But since any difference in the beginning and end conditions may lead to considerable error, which can only be equalized by excessive length of test, the "Rules" are intended to operate to the end that such errors are not in any way magnified by the method of test. Hence the detailed statement in the regulations.

27

Since in actual operation the fuel in the ash or the coal dust in the gas mains are hardly ever utilized, no correction should

be made for these on any trial. In order, however, to prevent the results from being influenced by insufficient cleaning of the producer, any fuel which falls out from above the grate during the cleaning period may be subtracted from the amount charged.

35

See explanation under 23.

37

A brake test of a large engine is in some instances not possible. and in any case a matter of considerable cost. In many cases, however, the larger gas engines are either direct connected to a generator or to some other power consumer, as a blowing cylinder. In the former case electrical measurements, from which the effective horse-power may be determined, are easily made. In the latter case the capacity guarantee will in most instances be based upon the performance of the power consumer, as for example the air compressed by the blowing cylinder. Outside of engines of this type, however; there still remain many cases in which it would be of the utmost value to have some means of determining the effective capacity, and it should not be forgotten that, even in the case of medium-sized machines, a braking of the engine at the place of erection is often, on account of local restrictions, very difficult. The problem has been solved for steam engines by assuming that the difference between the indicated horse-power at any load and the indicated horse-power at no load is the effective or useful horse-power. It is quite possible that in many cases this is not quite correct, but the method is very generally accepted and followed.

On account of the great overload capacity of the steam engine, a small error in this respect does not mean a great deal. But the case of the gas engine is quite different. The data on hand does not warrant the application of the same method to the gas engine, and the consequences of an erroneous conclusion are much more serious on account of the lack of overload capacity.

For these reasons one is compelled in some cases to omit the determination of the effective capacity altogether and to be content with the determination of the indicated power only. It is recommended in such cases that the mechanical efficiency be not

assumed too high and that any guarantees regarding fuel, etc., also be based upon the indicated horse-power.

It is sometimes possible to brake an engine on the test floor of the factory. The mechanical efficiency may thus be previously determined when it is known that no brake test can be made in the final place of erection.

39

The number of diagrams to be taken during any given test cannot be definitely specified. Much depends upon the length of test, and the decision may be left to the judgment of the experimenter.

It is, however, always recommended that a bundle of diagrams, instead of only one, be taken on every card. Thus a series of diagrams are obtained, while, if only a single diagram is taken, it is possible to hit upon the same diagram in the series a number of times. (See under extract 8.)

The work of fluid friction, that is, the lower-loop diagram, cannot be determined with certainty from the full indicator cards. It is best for that reason to ignore the loop when determining the positive work and to find the negative work from special weak spring diagrams.

48

The measurements of the quantity of lubricating oil used is of importance in smaller engines, because the fuel consumption can be favorably influenced by a copious supply of the lubricant.

49

If under low loads, only one end of the cylinder is allowed to work, the fuel consumption would be much lower. But since this is not generally done in operation, the results would be erroneous. If, however, the governor during operation shuts off the individual cylinders or cylinder ends, as the load drops, this is of course also permissible during a test.

CHAPTER XVII

THE PERFORMANCE OF GAS ENGINES AND GAS PRODUCERS

1. As indicated in the rules for testing, the very great majority of tests of engines are made to determine capacity and fuel consumption. In some special cases, as with engines driving generators, tests are also sometimes made of the regulation. These three tests together take care of what may be termed the commercial side of testing. All other tests are special in that they are not often executed in an acceptance test, but form in most cases the object of scientific or laboratory investigation.

Such investigations are in many instances very valuable, and have served to throw a flood of light on the somewhat complex cylinder actions of the internal combustion engine. It was thus found that the temperature of the cylinder walls, *i.e.*, the cooling water conditions, piston speed, ignition, proportion of mixture, compression, etc., all had a more or less marked effect upon engine performance. In the following the results obtained and the conclusions drawn by various experimenters concerning the effect of these various factors are briefly set forth.

The main bulk of this work has been done by Witz in France, Slaby and E. Meyer in Germany, and Burstall and others in England. In spite of the fact that none of these investigations are open to serious objection on the score of inaccuracy, the conclusions arrived at are not always in accord. This is undoubtedly due to the complexity of the cylinder actions, and the interdependence of the various factors entering the problem.

2. **Cooling Water Conditions and Piston Speed.** — It is reasonable to suppose that the higher the wall temperature of the cylinder, *i.e.*, the smaller the temperature difference between mixture and wall and the greater the piston speed, cutting down the time of exposure, the smaller the loss to the jacket.

The heat thus saved, however, may go in two directions: either

a greater thermal efficiency is shown, resulting in greater power developed for the same expenditure of heat, or the heat saved from going into the jacket is lost in the exhaust.

Witz, upon the basis of his experiments, comes to the former conclusion, and says "The action of the jacket is the great regulator of combustion phenomena." He summarizes his results as follows:

1. The efficiency increases with the piston speed and with the temperature of the surrounding walls.
2. The combustion of explosive mixtures is the more rapid the greater the speed of expansion and the hotter the cylinder walls.

The work of Slaby and of Meyer, however, seems to controvert these conclusions. Some of Slaby's tests show that while the gas consumption per horse-power hour decreases somewhat with an increase in the piston speed, there is an increase in the consumption with a rise of jacket water temperature. The following table of figures, quoted by Schöttler from some of Meyer's tests, illustrates the point that the heat saved from the jacket by higher piston speed may go to the exhaust, leaving the thermal return practically unaffected.

Ratio of Compression	R. P. M.	Mean Effective Pressure	Ratio Air to Gas	Heating Value of Charge	Work done by 1 B.T.U.	Exhaust Temp.	Heat Distribution in %		
		No. 1 sq. inch		B.T.U.	Ft. Lbs.	°F	Work	Jacket Water	Exhaust
2.67	187	54.3	7.11	18.5	140	1022	18.0	51.2	30.8
2.67	247	51.7	7.35	17.4	141	1137	18.1	45.6	36.3
4.32	187	69.3	7.43	17.0	190	867	24.4	53.8	21.8
4.32	247	65.2	7.40	16.8	184	992	23.7	49.5	26.8

In tests of this kind there are one or two simultaneous actions, not directly under control, which may serve to modify the final result and account in a measure for the discrepancy appearing in the results above discussed. An increase in the temperature of the walls or an increase in the piston speed both cause a decrease in the charge volume, the former by heating the incoming charge and decreasing the density of the mixture, the latter by increased friction loss in pipes and ports. The direct result of

this is that the effect of the action of the cylinder wall upon the leaner charge is proportionately greater. Thus the beneficial effect of greater piston speed may be partly annulled by the relatively stronger action of the walls. Further it is true that a smaller charge weight means a lower compression pressure, and the comparatively greater admixture of burned gases at high speeds causes a less rapid combustion. Both of these actions tend to decrease the efficiency. There are thus several antagonistic actions, and the final result is consequently in many cases quite problematical.

The *net* result of an increase in the cylinder wall temperature or of the piston speed, or both, in an existing machine, is certainly a decrease of maximum capacity for reasons already pointed out. Further, the effect of a variation in the temperature of the jacket water, while perhaps not quite so marked in engines using gas, is certainly quite noticeable in liquid fuel engines, especially those using kerosene or alcohol. It is quite possible in kerosene engines, by running the jacket too cold, to increase the oil consumption seriously by condensation of the oil vapor on the comparatively cold cylinder surfaces. The same holds true of alcohol. Thus hot walls are in such cases of undoubted benefit. The limits to temperature are, of course, decrease of engine capacity and danger of pre-ignition.

3. **Compression.** — The theoretical effect of increasing the compression, and the commercial limits to this increase have already been discussed in Chapter III. Much of the increased efficiency of blast furnace and producer gas engines as compared with illuminating gas and liquid fuel engines is directly due to the greater compressions that the former fuels can stand. The above table of Meyer's results gives some idea of the gain that can be made with illuminating gas by increasing the compression. With a compression ratio of 2.67, the average thermal efficiency was 18.05 per cent, with a ratio of 4.32 the average was 24.05 per cent, a gain of $\frac{24.05 - 18.05}{18.05} = 33$ per cent.

Another test made by E. Meyer * on a 10 horse-power engine, which was operated with illuminating and with producer gas, gave the following results:

* E. Meyer, Z. d. V. d. I., July 5, 1902.

Compression Ratio	Indicated Thermal Efficiency	
	With Illuminating Gas	With Producer Gas
4.98	27.1	24.4
4.59	26.5	23.2
3.84	24.8	21.5

Here again the beneficial influence of the higher compression is marked, although the gain is not so great as in the former case, owing to the smaller change in the compression ratio.

Other instances pointing to the same result can be adduced without difficulty. See the next table below, also by E. Meyer, who has done an immense amount of work in the investigation of gas engines. Bánki took advantage of the principle in his gasoline engine, in which, by using water injection, he could employ compression ratios similar to those used in producer gas work and realized thermal efficiencies fully equal to those obtained with the leaner power gases.

4. The Mixture. — The inherent advantage of the use of lean mixtures has already been shown in Chapter III. Burstall * on the basis of his tests for the British Institution of Mechanical Engineers, concludes that the thermal efficiency depends upon the correct choice of the mixture, and that the ratio of air to gas should increase with the compression. His results, however, do not definitely warrant the latter part of this deduction, although it finds some support in the above-mentioned tests by Meyer, † as shown by the following table:

Test No.	Ratio of Compression	Ratio Air to Gas	Gas Consumption Cu. Ft. per I. H. P.-hour
1	2.67	6.41	27.1
2		8.08	25.4
3	3.23	6.38	22.6
4		8.07	21.6
5	3.87	5.93	20.6
6		8.29	18.5
7	4.32	6.00	19.4
8		8.35	17.9

* Proceedings, 1898, p. 209. † E. Meyer, Z. d. V. d. I., 1899, p. 361.

This table shows that, whatever the ratio of compression, the gas consumption is less with the leaner mixtures. The cause for this, besides the theoretical reason, may possibly be found in the fact that with leaner mixtures the maximum temperatures in the cycle are lower than with rich mixtures, always assuming, of course, that the mixture contains no excess gas.

How the efficiency of an engine may be affected by careless setting of the fuel valve is well shown in some results quoted by Lucke.* The test cited is on a 10 I. H. P. Otto engine governed by hit and miss. The fuel used was carbureted water gas.

Gas Valve Number	Efficiency
9	16.0
8	16.5
7	18.0
6	19.0
5	10.0

It is quite evident from the figures that the last setting wasted a lot of unburned gas, while the leaner mixtures were inefficient, probably due to sluggish combustion. The same thing was noticed in a series of tests on German alcohol engines, in which it was found that the setting of the fuel needle valve had a pronounced effect upon the economy.

5. **Variation of Point of Ignition.** — The effect on the appearance of the diagram of varying the point of ignition has already been discussed. To get some idea of the influence of varying time of ignition on engine capacity and efficiency, the following figures are given. The first set † was obtained in determining the range of adjustment of the igniter gear on an 8″ x 12″ horizontal hit-and miss-engine, running 265 r.p.m. on natural gas.

* Lucke, Gas Engine Design.

† Obtained through the courtesy of Mr. A. B. Gould, of the Wellman, Seaver, Morgan Co., Cleveland.

Card No.	Crank Angle below horizontal at time of ignition	Max. brake load possible at 265 R. P. M.
8	35°	47 lbs. net
9	33°	47 " "
10	32°	47½ " "
11	24°	47 " "
12	23°	44½ " "
13	15°	44 " "

The accompanying indicator cards are shown, much reduced, in Fig. 17-1. The scale of spring was 160 pounds.

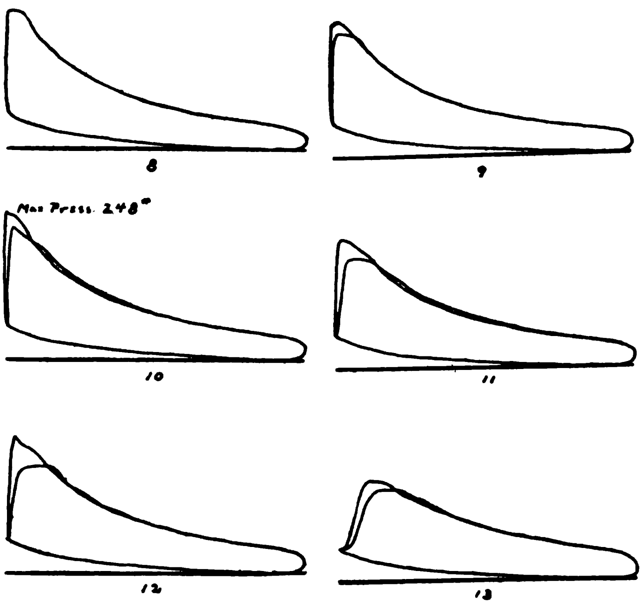


Fig. 17-1.

The second set of figures is due to Mr. J. R. Bibbins, and was published by him in the Michigan *Technic* for February, 1907. They are here reproduced by permission of the author, obtained through Mr. R. D. Day of the Westinghouse Machine Co. The tests were made on a 9 x 11″, 2-cylinder Westinghouse gas engine. The load was kept constant at about 70 B. H. P., the speed was held constant at about 300. r.p.m. The gas used was natural

gas with a constant lower heating value of 934 B. T. U. The point of ignition was changed by steps from dead center to 55 degree crank angle ahead of the center, that is, the spark was advanced to that extent. The following table shows the results:

Point of Ignition Degrees early	Load B. H. P.	R. P. M.	Gas per B. H. P. per hour cu. ft.	B. T. U. per B. H. P.-hour	Thermal Efficiency on Brake %	Relative Efficiency	Pressures Lbs. per sq. inch	
							Max.	Release
0	70.0	292	14.38	13410	19.0	.815	151	36
8	70.8	295	13.34	12470	20.4	.875	168	36
20	71.0	296	12.36	11530	22.1	.947	177.5	33.6
25	71.0	296	12.3	11490	22.2	.951	220.5	31.2
30	71.3	297	11.71	10940	23.3	1.000	252	31.2
35	71.3	297	12.03	11230	22.7	.972	252	28.8
45	71.2	296.5	12.40	11590	21.9	.942	379	28.8
55	70.0	292	15.74	14700	17.3	.742	437	24.0

The results of the test are shown graphically in Fig. 17–2. The best lead angle for the sparking gear appears to be between 30 and 35 degrees, in which this test agrees closely with the re-

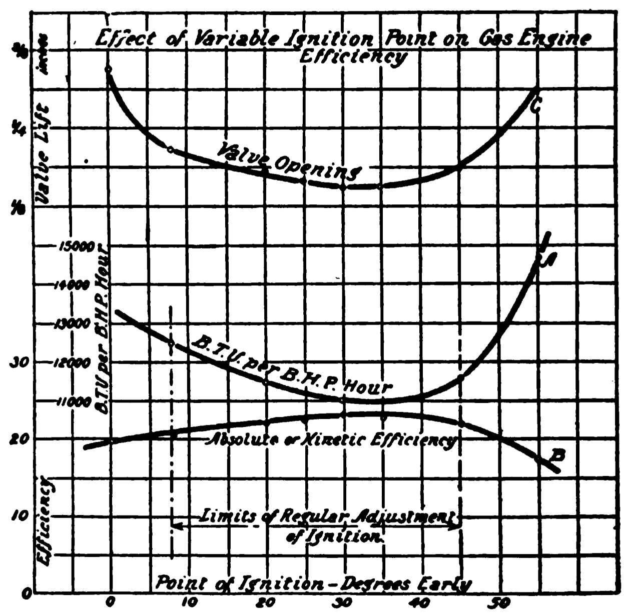


FIG. 17–2.

sults obtained by Mr. Gould on a similar engine. Fig. 17–3 shows the typical indicator card accompanying each igniter position.

6. Engine Economy depending upon Load. — As is the case in the steam engine, the efficiency of a gas engine decreases as the load decreases below the normal. The amount of this decrease varies in different engines, depending mainly upon the

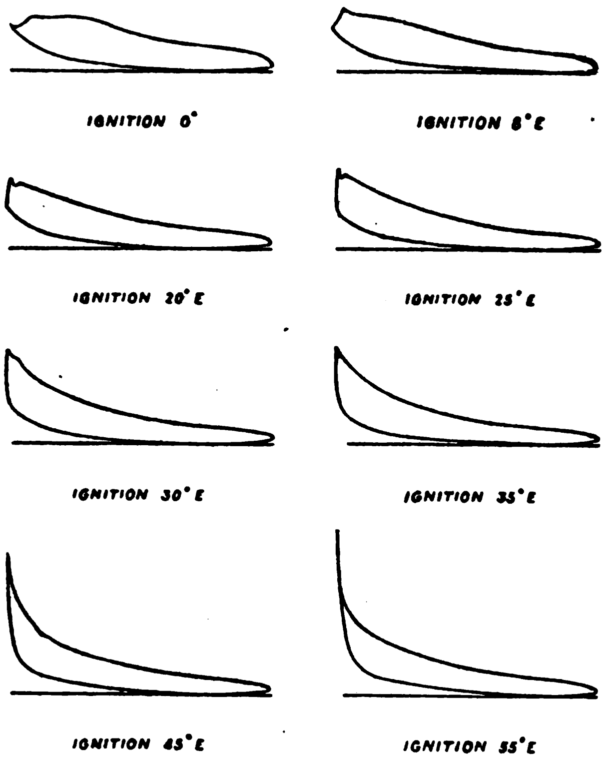


Fig. 17–3.

kind of fuel used and the system of governing employed. The following set of curves, Fig. 17–4, makes this clear. Regarding the range of load above normal, however, it is found that, while a steam engine generally shows a decrease in efficiency for overloads, the gas engine usually shows a greater efficiency at the maximum load than at the normal; in other words, just as long as a gas engine keeps up the normal speed under an increase in

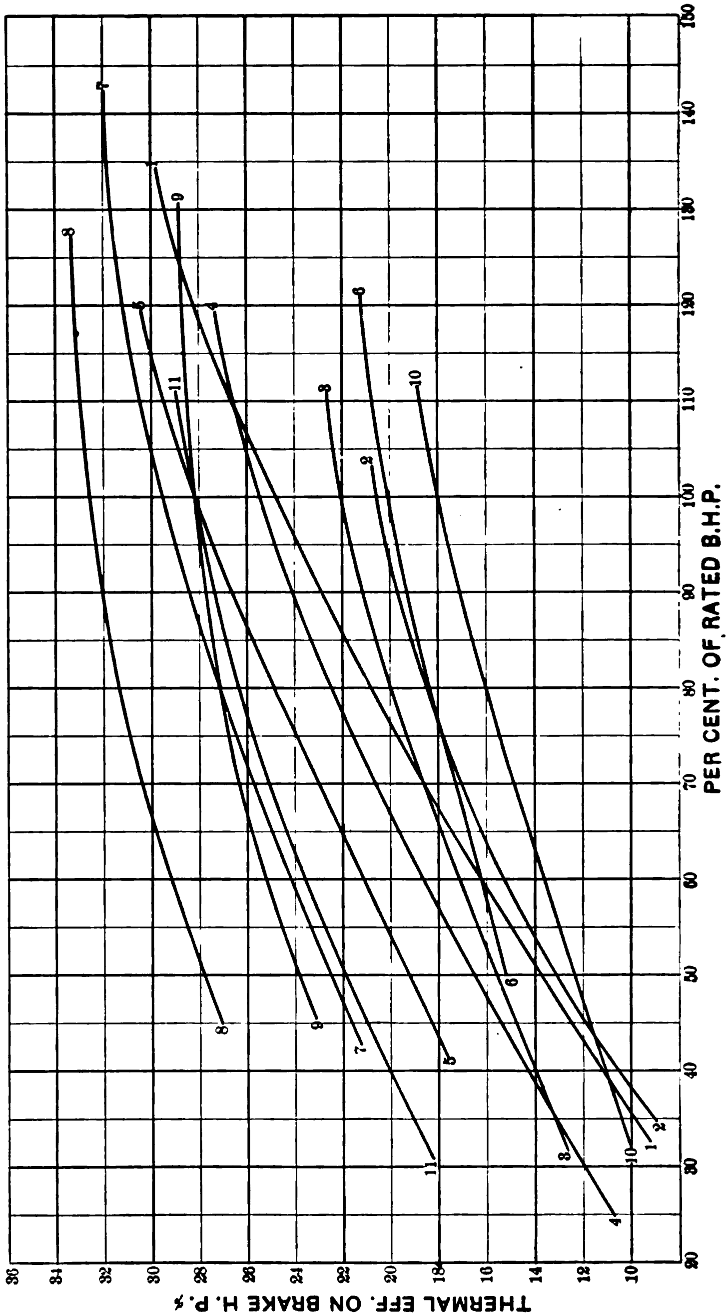


FIG. 174.

load the thermal efficiency will rise with the load. A few typical efficiency curves are shown in the figure. The data for these has been collected from various sources, as shown in the accompanying table:

Curve No.	Type of Engine	B.H.P.	R.P.M.	Fuel	Governing	Reference
1	Deutz, Single Cylinder	50	200	Ill. Gas	Throttling Gas.	E. Meyer, Z. d. V. d. I., 1898.
2	Westinghouse 3-cycl. Vert.	100	270	Natural Gas.	Throttling Mix.	Robertson, 1899.
3	Deutz.	450	—	Producer Gas.	—	Guarantee Fig. for entire plant. Josse, 1904.
4	Güldner, Single-cylinder.	35	220	Producer Gas.	Throttling	Güldner, 1906, for plant including generator.
5	Nürnberg.	1200	106	Blast-Furnace Gas.	Throttling Gas.	Riedler, Gross-Gas-Maschinen, 1905.
6	Swiderski, Single-cylinder.	15	235	Alcohol.	Hit& Miss.	E. Meyer, Z. d. V. d. I., 1903.
7	Deutz, Single-cylinder.	12	285	Alcohol.	Throttling.	"
8	Diesel.	70	158	Russian Kerosene.	Cut-off.	"
9	Diesel.	8	275	"	"	"
10	Hornsby-Akroyd	25	202	Kerosene.	Regulating Oil.	Robinson, 1898.
11	Bánki.	25	210	Gasoline with water injection.	Hit& Miss.	Jonas and Taborsky Z. d. Oest. Arch.& Ing V., 1900.

7. **The Heat Balance.** — Accounting for the heat furnished to a heat engine is called constructing the *Heat Balance.* In an internal combustion engine the heat supplied to the engine is that contained in the fuel furnished to the engine in a given time. For convenience all heat calculations are referred to some standard temperature, usually room temperature. It is usual to account for the heat in four separate items:

1. Heat represented in indicated work.
2. Heat carried off in the jacket water.

3. Heat lost in exhaust.
4. Heat loss due to radiation, conduction, etc.

Of these the first and second items admit of accurate determination, the fourth is nearly always found by difference between the sum of the other three items and the heat supplied. Item three is much more difficult of exact determination. Its calculation involves the determination of the weight of exhaust gases and of the specific heat of these gases at exhaust temperature.

The only accurate way to find the weight of the exhaust gases is by metering or otherwise determining the air supply to the engine. The weight of the exhaust gases is in all cases the weight of air plus the weight of the fuel.

There are two other ways sometimes employed, but either one can only give approximate results. One of these determines the charge weight from the piston displacement. This involves

(*a*) The volumetric efficiency of the suction stroke, and

(*b*) Some assumption as to the temperature of the charge at the end of the suction stroke.

It is possible to determine the volumetric efficiency with fair accuracy by means of a weak spring card, but the second point offers much more difficulty. The charge at the end of the suction stroke, taking the case of a gas engine, consists of a certain amount of air, of fresh gas, and of burned gases from the clearance. The entering temperature of air and of gas can be accurately determined, but the temperature of the mixture, as it enters the cylinder, changes, due to contact with the hot walls and to mixing with the clearance gases. Nothing is definitely known of the weight of the clearance gases, for although their pressure and volume are known, nothing is known of the temperature. Hence neither the wall effect nor that due to the clearance gases can be definitely gaged and all temperature computations therefore become approximate. The only positive way of determining the temperature at the end of the suction stroke is by actual measurement. This has been successfully done, but the apparatus is not such as could well be employed in ordinary testing.

It must be evident, therefore, that piston displacement measurement of the weight of the exhaust gases can be approximate only. To cite a case in point, the test of Brooks and Stewart, mentioned in Chapter V, showed an actual ratio of air to gas on

test of 6.63, while piston displacement computation showed 8.32.

The second approximate computation for the weight of the exhaust gases is based upon the exhaust gas analysis. The method of doing this has been explained in detail in Chapter VI. The trouble with this scheme lies in the difficulty of obtaining representative gas samples. But granting even that these are obtained, it is often found that computations based upon the analyses show an excess coefficient smaller than that really used. That is, the weight of exhaust gases so determined is less than the actual amount. Thus, Schöttler, in computation on some of Slaby's tests, shows in one case that, based upon analysis, the excess coefficient was 5.52, while in reality it was 6.2. Schöttler attributes this discrepancy to a change in the analysis due to a burning up of the lubricating oil, which is apt to increase the CO_2 content of the exhaust gas at the expense of the percentage of O. It has been shown that a variation in the supply of lubricating oil may change the fuel consumption under circumstances quite materially, and Schöttler's surmise is therefore probably correct. At the same time, however, any such effect must be more marked in the smaller machines, and the writer believes that, given representative samples of exhaust gas from anything but the small machines, in conjunction with accurate fuel analysis, a very close approximation to the actual weight of the exhaust gases can be obtained. At any rate the method should be more accurate than that based on piston displacement.

Finally, it should not be forgotten that even an accurate determination of the air supply still leaves open the question of the specific heat of the exhaust gases.

The heat balance of the four items above outlined is sometimes shortened to three by combining items 3 and 4 and determining their sum by difference. On the other hand, a balance of many more items can be drawn up. Thus each event of the cycle may be examined by itself and the heat and energy interchanges be determined. A very detailed balance of this kind is given in Schöttler's "Die Gasmaschine," p. 321. How far this question needs to be entered into depends altogether upon the importance of the test, but an effort should be made in every case to so arrange

the apparatus that at least a heat balance of the kind first discussed can be drawn up from the results of the test.

8. Results of Tests of Engines and Gas Producers. — By this term is here meant the results shown by engines or producers when operated under normal conditions and at or near normal load. The number of tests from which this data can be obtained is at the present writing quite large, and some notable collections of test data have been made. The most extensive is perhaps that contained in Appendix A of Brian Donkin's "Gas, Oil and Air Engines." This collection consists of eleven tables, arranged according to fuel used, containing in all some 280 tests. Another large collection is that found appended to Witz's "Moteurs à Gas et à Pétrole." Such collections are valuable as showing what has been done, and serve as a guide as to what may be expected from an engine under design or construction. The greatest care, however, should be exercised to see that only reliable data is incorporated.

Engine Tests. — Table I (pp. 544 and 545) contains a series of engine test data taken from various sources. The tests are arranged according to the kind of fuel used, this being the most logical way. In some cases not all the data is given in the original report. Wherever possible computations have been made to make the items complete. In many cases, however, not enough information is given to permit of this, and the record necessarily remains incomplete.

Tests of Producers and Producer Plants. — Table II, p. 546, gives some of the results obtained on tests of producers. In some cases the tests refer to producers only, giving no data regarding the engine used; in others the data is fairly complete for the entire plant.

In conclusion, Fig. 17–5 shows a set of curves drawn by C. H. Day in 1905 during an investigation on the economy of producer gas. The curves show the relation between cubic feet of gas used and brake horse-power for various kinds of gas. While it must be understood that they are approximate only, they represent a fair general average of what is accomplished to-day. Many plants show much better fuel consumption, but others show

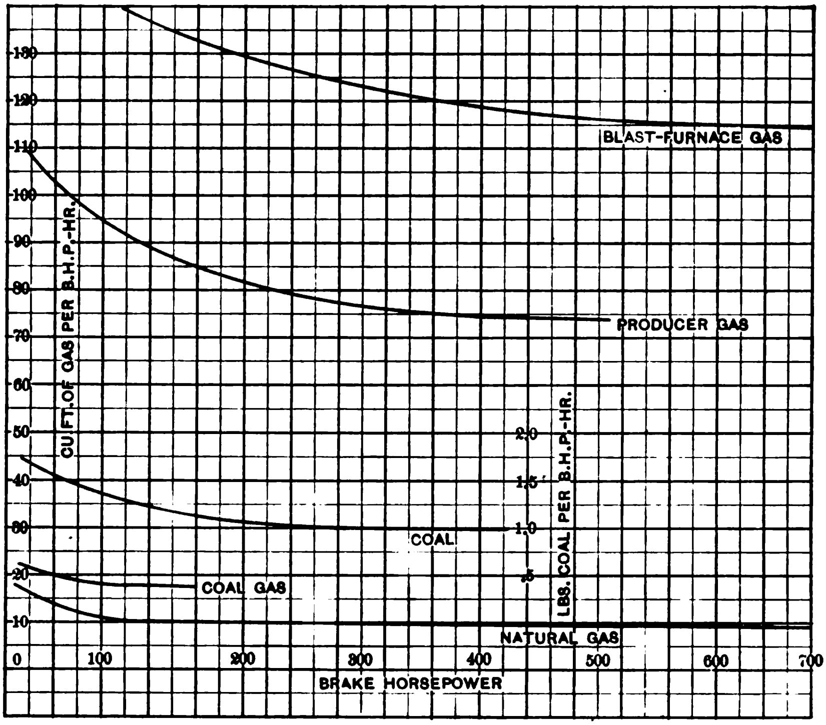


FIG. 17–5.

correspondingly worse. The curves also show in a measure the sizes up to which the various engines are built. Thus an illuminating gas engine of 150 horse-power is a large engine of its type. Producer plants in 1905 apparently were not built much larger than 500 horse-power, while natural gas and blast furnace gas engines were built exceeding 1000 horse-power, and for the latter gas 2000 and 3000 horse-power is not now out of ordinary.

TABLE I.

Kind of Fuel	Name of Engine	Dimensions and Other Data: Dia. of Cyl. Inch.	Stroke Inch.	No. of Cyls.	2 or 4-cycle	Single or Double Act.	Rated B.H.P.	R.p.m.
Illuminating Gas	Koerting	—	—	—	4	S	—	161
Illuminating Gas	Deutz	4.96	5.92	1	4	S	2	260
Illuminating Gas	Güldner	—	—	1	4	S	15	—
Illuminating Gas	Güldner	10	15.7	1	4	S	20	210.7
Illuminating Gas	Bánki	—	—	1	4	S	16	255
Illuminating Gas	Tangye	10	19	1	4	S	—	193.6
Illuminating Gas	Crossley	7	15	1	4	S	—	200
Illuminating Gas	Westinghouse	—	—	3	4	S	—	235
Natural Gas	Westinghouse	25	30	3	4	S	550	150
Natural Gas	Snow	25	48	4	4	—	—	—
Natural Gas	Otto	11.25	18	1	4	S	36	220.4
Natural Gas	Walrath	13	14	3	4	S	75	253.3
Producer Gas	R. D. Wood	25	30	2 (Tandem)	4	S	300	149
Producer Gas	Westinghouse	—	—	3	4	S	—	235
Producer Gas	Crossley	26	36	—	4	S	—	152.4
Producer Gas	Koerting	21.6	37.7	1	2	D	—	101
Producer Gas	Deutz	—	—	—	4	—	—	161.6
Coke Oven Gas	Oechelhäuser (Borsig)	65	37.5	1	2	2 pistons in 1 cylinder	500	110.6
Mond Gas	Crossley	16.9	24	2 (Opposed)	4	S	—	162
Mond Gas	Crossley	26	36	2 (Opposed)	4	S	400	148.5
Blast Furnace Gas	Nürnberg	33.5	43.4	2 (Tandem)	4	D	—	105.6
Blast Furnace Gas	Berlin-Anhalt	16.95	27.5	1	4	S	60	160.6
Blast Furnace Gas	Cockerill	51.2	55.2	1	4	D	—	94.57
Gasoline	Fairbanks	6.5	9	1	4	S	7	300
Gasoline	Springfield	6.5	12	1	4	S	6	230.4
Gasoline	Lozier	5	6	1	2	S	5	513
Gasoline	Westinghouse	5.75	8	2	4	S	10	289.7
Gasoline	Bánki	—	—	1	4	S	—	209.1
Gasoline	Daimler	3.56	5.11	4	4	S	16–20	400
Gasoline	Daimler	3.56	5.11	4	4	S	16–20	600
Gasoline	Daimler	3.56	5.11	4	4	S	16–20	1000
Kerosene	Diesel	10.23	16.16	1	4	S	—	186.6
Kerosene, Russian	Diesel	15.75	23.65	1	4	S	70	158.8
Kerosene, Russian	Diesel	6.65	10.60	1	4	S	8	270.3
Kerosene	Priestman	10.9	14.1	1	4	S	—	172
Kerosene	Grob & Co.	9.07	9.07	1	4	S	8	266
Kerosene	Swiderski	10	10	1	4	S	—	249
Kerosene	Koerting	6.9	10.82	1	4	S	—	222
Kerosene	Blackstone	7	12	1	4	S	—	240
Kerosene	Stevenson	9.5	18	1	4	S	—	200
Kerosene	Hornsby	8.2	14	1	4	S	—	213.8
Kerosene	Hornsby	14.5	17	1	4	S	25	202.6
Crude Oil	Diesel	—	—	—	4	S	225	169.1
Alcohol, 86.1 vol.%	Deutz	8.35	11.8	1	4	S	12	276.9
Alcohol, 86.1 vol.%	Marienfelde	9.95	15.75	1	4	S	14	197.6
Alcohol, 86.1 vol.%	Koerting	6.16	9.95	1	4	S	6	307.3
Alcohol, 87 vol. %	Bánki	—	—	1	4	S	20	225

H.P. on Test		Lower Heating Value of Fuel B.T.U. per		Heat Distribution, %				Mech. Eff. %	Thermal Eff. on B.H.P. %	References and Remarks
I.H.P.	B.H.P.	Lb.	Cu. ft.	Work I.H.P.	Jacket	Exhaust	Rest			
—	108.1	—	500	—	—	—	—	—	28.22	De La Vergne Machine Co., Cat.
2.30	1.72	—	572	21.5	50.4	25.0	4.1	75	16.1	Wimplinger, Zeitschrift d. V. d. I. Sept. 8, 1906.
—	—	—	580	40.5	—	—	—	—	—	Dubbel, Z. d. V. d. I., Nov. 3, 1906.
35.9	—	—	—	42.7	33.2	24.1		—	—	Test by Schröter, Z. d. V. d. I., June, 1904.
—	17.03	—	—	—	27.97	—	—	—	31.0	Schimanek, Z. d. V. d. I., Jan. 17, 1903.
28.6	25.4	—	609	—	—	—	—	85	24.5	Witz, 1902.
14	12	—	680	—	—	—	—	86	22	Clerk, 1894.
196.5	147.5	—	597.4	35	—	—	—	76	26.7	Ballinger & Hunt, Sibley College Thesis, 1904.
618	550	—	1000	27.1	—	—	—	89	24.1	J. R. Bibbins, A.I.E.E., Dec. 1903.
736.7	594.5	—	1175(?)	29.4	—	—	—	80.7	23.7	Hastings & Parker, Sib. Coll. Thesis, 1901.
36.3	28.8	—	1086	20.2	—	—	—	79.5	16.1	Hunting, Sib. Coll. Thesis, 1902.
86.7	78.7	—	1041	27.1	49.5	23.4		78.7	21.3	Geer & Vanelain, Sib. Coll. Thesis, 1902.
242	169	—	145	24.4	25.0	50.6		70	17.1	Goldsmith & Hartwig, Sib. Coll. Thesis, 1905.
173.6	124.8	—	123.8	33	—	—	—	71.8	23.7	Ballinger & Hunt, Sib. Coll. Thesis, 1904.
377.9	313	—	90.13	—	—	—	—	83	21.78	Humphrey, Inst. Mech. Engs., 1900.
481	341	—	129.5	—	—	—	—	71	24.1	E. Meyer, Jour. für Gas Belenchtung, 1900.
81	70.4	—	113	—	—	—	—	87	29	E. Meyer, 1903.
765 (net)	628 (Blowing Cyl).	—	376	33.4	—	—	—	82.1	27.5	E. Meyer, Z. d. V. d. I., Feb. 1905. Test No. VIIb.
141.6	—	—	135	29.1	—	—	—	—	—	Humphrey's Test, Schöttler, Z. Jan. 1902.
440	364	—	138	28.6	—	—	—	83	23.7	
1427	1186	—	88	33.9	—	—	—	83.1	28.2	Riedler, Gross-gasmaschinen, p. 158.
79.5	54.5	—	102	—	—	—	—	—	26.2	E. Meyer, Z. 1899, p. 448.
786.16	575	—	99.5	—	—	—	—	—	20.4	Hubert, March, 1900.
11.28	7.23	18200	—	20.5	—	—	—	64	13.15	Elwood, Ford, Garrow, Sib. Coll. Thesis, 1907.
9.63	6.16	20078	—	15.8	—	—	—	64	10.2	Keeley & Spier, Sib. Coll. Thesis, 1900.
6.28	4.92	18520	—	19.8	—	—	—	78.9	15.7	Bayne & Speiden, Sib. Coll. Thesis, 1902.
9.19	5.58	—	—	28.5	—	—	—	60.7	17.3	Glasgow & Powley, Sib. Coll. Thesis, 1902.
—	26.4	—	—	—	—	—	—	—	28	Jonas & Taborsky, Z. Oest. Arch.&Ing. Verein, p. 512, 1900.
—	—	17500	—	19.3	—	—	—	86.1	16.6	Prof. B. Hopkinson, Cambridge, 1906.
—	—	17500	—	22.0	—	—	—	85.4	18.8	
—	—	17500	—	24.2	—	—	—	79.8	19.3	
30.46	20.81	18604	—	37.7	—	—	—	68.3	25.8	Cat. American Diesel Engine Co.
88	69.6	18610	—	40.3	—	—	—	79.1	31.9	E. Meyer, Z. d. V. d. I., May, 1903.
11.19	8.6	18610	—	35.8	—	—	—	77.0	27.6	
10.69	10.18	19800	—	—	—	—	—	95.2(?)	13.4	Hartman, Z. d. V. d. I., 1895, p. 586.
11.68	9.22	19800	—	—	—	—	—	79	13.6	Hartman, Z. d. V. d. I., 1895, p. 616.
—	10	19600	—	—	—	—	—	—	15.8	Schöttler, die Gasmaschine, p. 207.
—	4.15	19600	—	—	—	—	—	—	9.9	
—	3.05	18000	—	—	—	—	—	—	19.7	Engineering, Vol. 88, 1899.
—	12.44	18000	—	—	—	—	—	—	9.2	
6.07	4.95	18600	—	16	29	55		81.5	13	Robinson, 1893, Gas & Pet. Engines, p. 702.
31	26.74	18870	—	21	50	29		86.0	18	Robinson, 1898, Gas & Pet. Engines, p. 710.
—	249.7	19460	—	—	—	—	—	—	28.1	Kimberley & Clark Paper Co., Kimberley, Wis., Power, Oct. 1906.
—	16.8	9900	—	—	—	—	—	—	31.6	E. Meyer, Z. d. V. d. I., April, 1903.
—	19.77	9900	—	—	—	—	—	—	32.7	
—	7.39	9900	—	—	—	—	—	—	21.8	
—	32.13	9700	—	—	32.1	—	—	—	30.18	Schimanek, Z. d. V. d. I., Jan. 17, 1903.

TABLE II. Tests of Gas Producers

Make of Engine	Make of Producer	Type of Producer	Kind of Fuel	Heating Value of Fuel B.T.U.	Engine: Dia. Cyl. Inch.	Engine: Stroke Inch.	Engine: R. p.m.	Horse-Power: Indicated	Horse-Power: Brake	Producer Efficiency %	Fuel, Lbs. per Hour: I.H.P.	Fuel, Lbs. per Hour: B.H.P.	Heating Value of Gas B.T.U. per cu. ft.	References and Remarks
R. D. Wood	Taylor	Pressure	Anthracite	10240	25	30	149	242	169	79.4 (cold gas)	1.28	1.83	145	Goldsmith & Hartwig, Sibley College Thesis, 1905.
Güldner	Güldner	Suction	Anthracite	13630	—	—	220.8	44.8	35.6	—	.692	.875	—	Zeitschrift des Vereins deutscher Ingenieure, June 25, 1904.
Mees	Pintsch	Suction	Coke	13270	11	15.7	220	25.76	20.2	—	.766	1.09	—	Z. d. V. d. I., June 17, 1905.
Mees	Pintsch	Suction	Anthracite	14620	11	15.7	220	26.30	21.48	—	.732	.895	—	
Campbell	Campbell	Suction	Scotch Anth. Pea Coal	—	9.5	18.0	192.4	23.73	20.44	—	—	.93	—	Trials of Suction Gas Plants, Highland and Agricultural Soc. of Scotland, Power, Feb., 1906.
Campbell	Campbell	Suction	"	—	7	12	232.5	9.66	8.25	—	—	1.22	—	
Crossley	Crossley	Suction	"	—	8.5	20	189.2	18.25	15.35	—	—	.77	—	
Ind. Eng. Co. Manchester	Ind. Eng. Co., Manchester	Suction	"	—	8.5	14	204.9	—	9.72	—	—	1.13	—	
National	National	Suction	"	—	7	15	219.80	11.88	9.74	—	—	.84	—	
National	National	Suction	"	—	10	18	193	25.54	20.57	—	—	.80	—	
Tangye	Tangye	Suction	"	—	7	16	224.4	10.86	8.34	—	—	1.25	—	
Tangye	Tangye	Suction	"	—	10	19	197.9	23.45	19.98	—	—	.83	—	
Nürnberg, Single Act.		Suction	Anthracite	—	—	—	—	134.4	107.4	80	.78	.975	—	Test at Royal Foundry of Wurtemberg; Mathot in Power, Feb. 07
Nürnberg, Single Act.		Suction	Coke	—	—	—	—	137.5	110.0	75	.93	1.16	—	Test in Post Office, Hamburg
Tangye		Suction	Scotch & Welsh Anth.	—	11	20	186	36.34	29.46	—	—	.709	—	R. E. Mathot in Power, May, 1906.
Westinghouse, 500 H.P.	Loomis-Pettibone	—	Pa., Bit. & Pocahontas Slack	—	—	—	—	—	—	—	—	1.42	—	Includes stand-by loss. Norton Plant, Worcester, Mass., Power, July, '07.
Westinghouse, 250 H.P.	Taylor	Pressure	Colorado Bit.	9767	—	—	—	—	—	—	—	1.95	—	Data from Tests, collected by J. I. Wile, Power, April, 1906.
Stockport, 250 H.P.	Wilson	Pressure	Bituminous	—	—	—	—	—	—	—	—	1.40	—	
Campbell, 280 H.P	Dowson	Pressure	Bituminous	—	—	—	—	—	—	—	—	1.12	—	
Westinghouse, 3000 H.P.	Dowson	Pressure	Anthracite	—	—	—	—	—	—	—	—	1.20	—	
Deutz, 250 H.P.	Deutz	Suction	Anthracite	14600	—	—	—	—	—	—	—	.744	—	
Crossley, 90 H.P.	Crossley	Suction	Coke	—	—	—	—	—	—	—	—	.91	—	
Crossley, 300 H.P.	Crossley	Suction	Anthracite	11370	—	—	—	—	—	—	—	.975	—	
Westinghouse, 225 H.P.	Taylor	Pressure	Colorado, No. 1 Lignite	—	19	22	200	—	235.4	64.2	—	1.55	—	Test No. 15, U. S. Geological Survey Coal Testing Plant, St. Louis, Mo.
Letombe	Letombe	Suction	—	—	23.6	31.5	135	—	290	—	—	.81	—	Report by A. Witz.

CHAPTER XVIII

COST OF INSTALLATION AND OF OPERATION

1. **Cost of Producers and Engines.** — With the approach to standardization of the design of the internal combustion engine there has also come a tendency toward standard prices. The figures available, however, for anything but small engines are not plentiful. This is no doubt due to the fact that large installations are as yet not numerous and information is correspondingly scarce. The writer has been able to glean the following data from current literature:

PRODUCERS. — The *Gas Engine*, June, 1906, gives the following figures for producers:

Pressure Producers, erected complete but not including freight, \$25 per horse-power for hard coal, and \$27 per horse-power for soft coal producers. These figures are for sizes in the neighborhood of 300 horse-power. The cost is distributed approximately as follows:

25 per cent in the producer.
15 per cent in the scrubber.
30 per cent in the gas holder.
30 per cent in piping and fittings.

The increased cost of the bituminous coal plant is mainly due to greater cost of washing apparatus.

Suction Gas Plants will, on a rough estimate, cost only about one-half the above. The reasons for this are greater capacity of producer and the elimination of the gas holder. One maker gives the following approximate figures for the price of suction gas plants from 25 to 250 horse-power.

$$\text{Price per horse-power} = \$11 + \frac{400}{\text{H.P.}}$$

Thus a 25 horse-power plant would cost \$37.00 per horse-power, and one of 250 horse-power \$12.60 per horse-power. In the writer's experience practice agrees fairly well with these figures.

Engines. — The cost of stationary gas engines at present seems to range from about $70 per horse-power for very small engines to in the neighborhood of $25 per horse-power for large ones. Recent quotations on producer-gas engines for instance showed the cost to be about $55 per horse-power for a 10 horse-power engine, and $36 per horse-power for a 50 horse-power machine. The latter is, however, a comparatively low price for an engine of this size. The price of launch engines is frequently lower than this, from $50 per horse-power for the small engine to about $20 per horse-power for an engine of 25 to 30 horse-power. The cost of Diesel engines runs somewhat higher, varying probably between $50 and $90 per horse-power, depending upon size. None of these figures include cost of erection.

Suction Gas Plants. — Recent quotations for plants of this type have been for a 10 horse-power plant, which is below the ordinary commercial size, $90 per horse-power for a 50 horse-power plant $51 per horse-power and again $61 per horse-power, and for a 75 horse-power plant $53 per horse-power.

2. **Cost of Erection.** — The cost of erection varies largely with the size of the foundations. Regarding the latter there seems to be no definite proportionality between it and the size of the engine. In general, of course, vertical engines do not require as large or as heavy a foundation as horizontal engines, but the condition of the ground has a great deal to do with the final cost. Güldner * estimates that from 15 to 20 cubic feet of foundation per brake horse-power of engine should be sufficient in the ordinary case if the ground is safe. For large machines, however, this may even be as low as 8 cu. ft. per B. H. P. The total cost of erection and starting should not exceed about 3 per cent of the purchase price in the case of gasoline and gas engines, and should not be over 5 per cent in the case of producer gas plants. These figures agree fairly well with the estimates made by H. A. Clark in 1903.†

3. **Piping and Auxiliaries.** — This item is apt to vary within wide limits, especially in producer gas plants, where much depends upon the location of the producer plant with reference to

* Verbrennungsmotoren, p. 433.

† H. Ade Clark, The Diesel Engine, Proceedings of Mechanical Engineers, 1903.

the engine. In a normal case, 5 per cent of the cost price of the engine or plant should cover the cost. This, however, does not include the cost of an air compressor system if this is required to start the plant.

4. **Floor Space and Buildings.** — The cost of buildings is intimately connected with the floor space required by the plant, but outside of this nothing very definite can be said of the cost of the building, since everything depends upon the construction used. The cost based on the square foot of floor space may vary from about $1.50 for a plain brick building with steel roof, to $2.75 for a modern steel concrete structure. Among the figures available are the following. A. H. Clark, in the article already mentioned, estimates the cost of building as follows:

Type of Engine	Size of Engine B. H. P.	Cost of Engine, Producer and Accessories	Cost of Building	Ratio $\frac{\text{Cost of Bldg.}}{\text{Cost of Plant}}$
Producer Gas * . .	35	$2,360	$ 640	.27
Diesel	35	2,860	440	.16
Producer Gas . . .	80	4,400	930	.21
Diesel	80	4,400	685	.16
Producer Gas . . .	160	7,450	1,150	.16
Diesel	160	8,300	880	.11

Güldner, in making a similar estimate, combines the cost of building and of necessary foundations for the engine and producer. Expressing this combined cost in per cent of the cost of plant and accessories, the following table shows the results:

Type of Plant	Horse-power of Plant												
	5	10	15	20	25	30	40	50	100	125	150	175	200
Illuminating Gas Engine	27	22	20	19	18	17	16	15	13	13	13	13	13
Suction Gas Plant Complete	30	26	24	22.5	21	19.5	19	18	15.5	15	15	15	14.5

Turning next to the floor space required, there is considerable data available in the literature published. Collaborating the

* Pressure producer.

figures given by manufacturers for the floor space required for horizontal single-cylinder gas, gasoline and oil engines, and for suction gas producers, we obtain Curves I and III of Fig. 18–1.

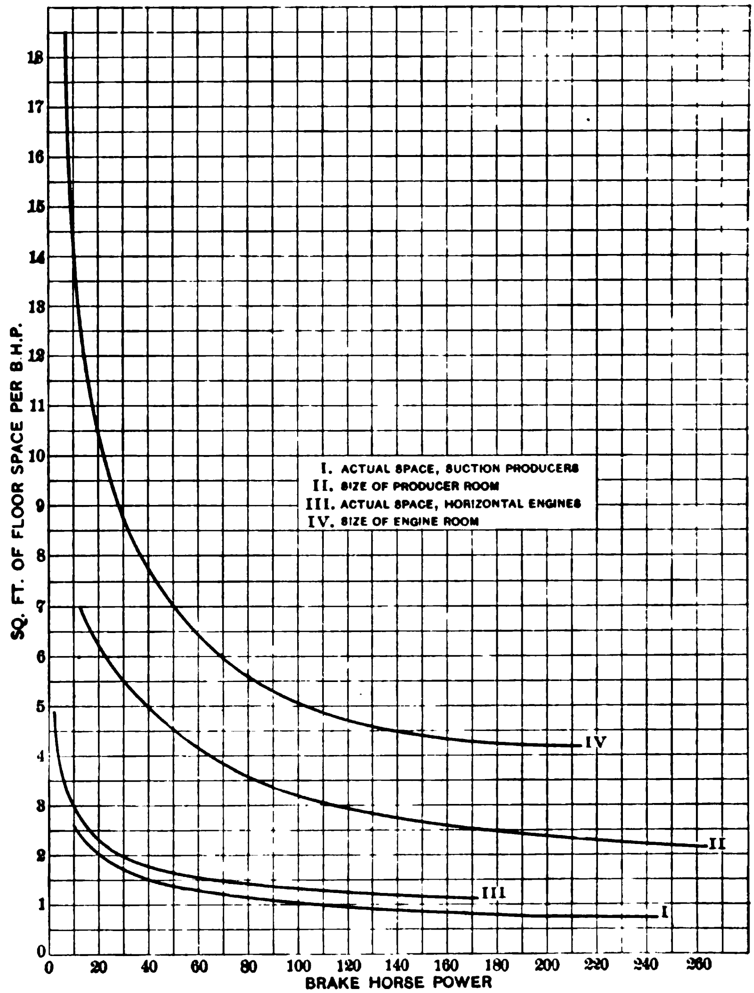


FIG. 18–1.

It should be remembered in connection with these curves, however, that they represent average figures only and they can therefore give only approximate results. For example, a 20 B. H. P.

suction gas producer will, according to Curve I, occupy approximately $20 \times 2 = 40$ sq. ft. of floor space, say a space $9 \times 4\frac{1}{2}$ ft.; similarly, a 200 B. H. P. installation would require $200 \times .8 = 160$ sq. ft.

In like manner Curve III gives the approximate floor space for horizontal engines. Thus a 5 B. H. P. gasoline engine of standard make would probably require $5 \times 4 = 20$ sq. ft., say a space 3 ft. by 6 ft.; while a 150 B. H. P. producer or illuminating gas engine calls for $150 \times 1.5 = 225$ sq. ft. The curve does not include multi-cylinder engines because the available figures for this type of engine are very erratic. For the same reason no curve could be drawn for vertical engines.

While Curves I and III represent the space actually occupied by the plant, they give no idea of the size of the producer or engine room required. In the producer room space must be left for the proper cleaning of the producer, etc., and in the engine room space for dismantling, etc. On this point the only information that seems available is that given by Güldner in his estimates, the results of which are plotted in Curves II and IV of Fig. 18–1. The former shows the approximate room allowance for suction gas apparatus, the latter for horizontal single-cylinder engines, based on one brake horse-power in each case.

The data available for *pressure producers* is not so extensive as that for suction producers. In the former plants much depends upon the size of the gas holder. Outside of this the space required for suction and pressure producers should not vary greatly. The figures at hand seem to indicate that pressure producer plants, including the gas holder, of from 30 to 40 horse-power occupy 75 per cent more space than suction plants of like capacity. As the power increases the difference grows less, a plant of 100 horse-power requiring apparently only about 50 per cent more space, and one of 175 horse-power only 20 per cent more.

The *minimum head room*, *i.e.*, the height of the room required for either type of plant, seems to vary from 12 ft. in the small 20 horse-power plant to 18 ft. in the large 250 horse-power producer.

In Power for April, 1907, L. L. Brewer published the following table of "Practical Data for Modern Gas Engines." The table gives in column 13 figures for the floor space required by engines larger than those so far discussed. The letters in column 7 have

the following meaning: s.c., single cylinder; tw., twin; td., tandem; d. td., double tandem.

1	2	3	4	5	6	7	8	9	10	11	12	13
Brake Horse-Power	Revolutions per Minute	Builder	No. Cylinders	Strokes per Cycle	Single or Double Acting	Cylinder Arrangement	Weight of Engine without Fly-wheel, Lbs.	Weight of Fly-wheel for Blowing and Pumping Service, Lbs.	Weight of Fly-wheel for Dynamo Driving	Weight of Engine and Fly-wheel (9) per B. H. P.	Weight of Engine and Fly-wheel (10) per B. H. P.	Sq. ft. Floor Space per B. H. P.
100	150	Cockerill	1	4	s.	sc.	45,000	9,000	21,100	540	661	2.05
200	105	Cockerill	1	4	s.	sc.	83,000	25,000	58,500	540	706	1.81
250	150	Cockerill	2	4	s.	td.	65,000	10,000	23,400	300	353	1.24
300	120	Deutz	1	4	s.	sc.	83,500	35,000	81,800	295	551	2.07
300	120	Deutz	2	4	s.	tw.	101,000	14,000	32,800	383	447	1.52
300	140	Deutz	4	4	s.	d. tw.	110,000	3,500	8,200	484	394	1.32
600	80	Cockerill	1	4	s.	sc.	207,000	100,000	234,000	512	734	0.99
600	130	Cockerill	2	4	s.	td.	185,000	46,000	107,500	386	487	1.13
600	110	Oechelhäuser	1	2	s.	sc.	143,000	48,000	112,000	318	425	1.23
600	130	Deutz	2	4	s.	tw.	158,000	28,000	65,500	310	371	1.67
600	130	Deutz	4	4	s.	d. tw.	189,000	7,000	16,400	327	342	1.08
600	110	Koerting	1	2	d.	sc.	136,500	18,000	42,200	258	297	1.11
750	90	Nürnberg	1	4	s.	sc.	297,000	115,000	26,900	560	538	1.03
1200	80	Cockerill	2	4	s.	td.	365,000	95,000	222,000	383	488	0.68
1200	130	Deutz	4	4	s.	d. tw.	354,000	14,000	32,800	307	322	1.01
1200	120	Nürnberg	4	4	s.	d. tw.	280,000	16,000	37,400	246	264	0.94
1200	110	Oechelhäuser	2	2	s.	tw.	260,000	16,000	37,400	230	248	0.9
1200	110	Koerting	2	2	d.	tw.	250,000	4,500	10,500	212	218	0.9
1400	110	Cockerill	2	4	d.	td.	374,000	8,000	20,000	170	164	0.42

It will be noted from column 6 that there is only one double-acting four-cycle engine in the list. The large engines cited are all single-acting twin or tandem, or double engines. It would naturally be assumed that the later types of large four-cycle engines, which are nearly always double-acting tandem, or double-acting twin tandem machines, should show a decrease in the floor space required as compared with the figures in column 13. An indication of the saving in both weight and floor space that may be effected by adopting the double-acting principle is given by the figures for the 1400 B. H. P. double-acting tandem Cockerill engine in the last line of the table. The saving there shown is remarkable and the writer has not been able to check it in the case of other engines the design of which has been changed from the twin or double-twin single-acting to the double-acting tandem. The reason for this probably is that opposed single-acting engines usually work without a cross-head,

while double-acting cylinders demand the use of the same. Hence the total length of engine is not changed materially.

5. **Cost of Operation.** — The total cost of operation consists of the following items:

(*a*) Interest on capital.

(*b*) Depreciation of Plant and Buildings.

(*c*) Insurance.

(*d*) Fuel Cost.

(*e*) Cost of Cooling Water.

(*f*) Lubricating Oil and Waste.

(*g*) Attendance.

(*h*) Maintenance and Repairs.

Of these, the first three items are usually called the *fixed charges*, and the last five the *operating* or *works cost.* The sum of fixed charges and works cost is the *total operating cost.*

(*a*) Interest on Capital. The usual allowance in this country for interest on capital is 6 per cent.

(*b*) Depreciation of Plant and Buildings. There is little doubt that as far as gas producers are concerned, the wear and tear on this part of the plant is much less than it is on a boiler plant of the same capacity. On the other hand, the stresses in a gas engine are generally higher throughout than those in a steam engine of the same power, and this naturally leads to a shorter life and consequently somewhat higher allowance for depreciation. Taking it altogether, an allowance of from 7 to 10 per cent of the capital outlay for the power plant should cover depreciation, the lower figure for a producer plant, the higher where engines alone are concerned.

Depreciation on the building should not exceed 2 to 3 per cent of the building cost. In case the building is rented, the rent instead of depreciation should be charged against the plant.

(*c*) Insurance. This item is usually very small as compared with the rest and is therefore in most cases neglected.

(*d*) Fuel Costs. This item is very often used as the sole criterion of the economic status of a plant, but in many cases this is not true. Where the cost of fuel is very high, as for instance where illuminating gas or gasoline are used, the fuel cost usually forms the major part of the operating expenses. But where the cheaper fuels are used or the plant is of high efficiency, as is often

the case in producer plants, an analysis will show that what are usually considered the "incidental" expenses in many cases far outweigh the fuel cost.

The fuel cost varies with the load on the engine. The figures given by manufacturers usually represent the best figures obtained at maximum load, but such conditions rarely obtain in practice. At any rate it would not be safe to base computations upon such "parade" figures. Haeder * considers that the economy figure given by an engine at $\frac{7}{10}$ maximum load, which is equal to 85 per cent of rated load if an over-capacity of 20 per cent is assumed, is the best figure on which to base computations. The same writer gives the following table for the variation of the fuel consumption with load. The third line in this table gives the same information regarding a steam plant:

Gas Engine	Variation of Fuel Consumption with Load									
	Max. Load	.9	.8	.7	.6	.5	.4	.3	.2	.1 of Max. Load
Hit-and-miss regulation	1	1.03	1.08	1.14	1.23	1.35	1.50	1.8	2.5	3.0
Throttling regulation .	1	1.06	1.13	1.21	1.33	1.50	1.75	2.2	3.0	5.0
Steam plant	1.1	1.05	1.02	1.0	.96	1.02	1.07	1.15	1.30	1.60

The table is based upon the assumption that the gas engine operates normally on $\frac{7}{10}$ of its maximum load, and this is put equal to the normal load on the steam engine. It is interesting to note the superiority of hit-and-miss regulation, as far as economy is concerned, over the other method of governing, and the superiority of the steam engine over both as regards small variation in economy over a wide range of load.

Gulder † makes a similar estimate with the following results:

INCREASE OF FUEL CONSUMPTION WITH DECREASING LOAD

Approximate Load	.75	.66	.50	.33	.25	of Normal Load.
Illuminating Gas Engine.	10	20	35	60	90	per cent higher fuel consumption than at full load.
Suction Gas Engine	20	30	50	75	100	
Diesel Oil Engine........	10	20	30	55	80	

* Haeder, Die Gasmotoren.

† Güldner, Verbrennungsmotoren, p. 427.

Thus, for example, if an illuminating gas engine uses 20 cu. ft. of gas per B. H. P. hour at full load, we may except it to use 1.35 × 20 = 27 cu. ft. per B. H. P. hour at half load, and 1.9 × 20 = 38 cu. ft. per B. H. P. hour at quarter load.

The anthracite coal consumption of suction gas plants for various sizes of plants and for different loads is given by Haeder in the following table:

Consumption of Anthracite in Suction Gas Plants in Pounds per B. H. P.-hour

Max. Capacity B. H. P.		14	40	70	100	140	210	420
Normal Load B. H. P.		10	30	50	70	100	150	300
Consumption pounds per H. P.-hour based on..	B. H. P.$_{max.}$	1.21	1.0	.93	.90	.88	.86	.84
	.7 B. H. P.$_{max.}$	1.45	1.19	1.12	1.10	1.06	1.03	1.01
	.5 B. H. P.$_{max.}$	1.80	1.47	1.38	1.34	1.32	1.28	1.25

It is evident from a study of the figures in the above tables that the probable average load at which a prospective plant will operate plays an important part in the estimation of fuel costs, a point which should not be lost sight of.

We find in the engineering literature of the day considerable information regarding fuel consumption and fuel costs. The cost figures given belong to one of two classes: either they are based upon assumptions regarding both consumption of fuel and cost of unit weight of the fuel, or the figures are obtained in actual operation. It goes without saying that the latter class is much more valuable than the former. In the case of assumed costs and consumptions, comparisons are usually made between steam and gas power under various conditions of operation. Such computations are interesting because they show what might be realized in each case, but in attempting to practically apply the information the greatest attention should be paid to the conditions assumed. As regards actual consumption figures, the tables of the previous chapter give considerable information regarding efficiency of operation. From this data it should be easy to compute the fuel costs as soon as the local cost of unit weight is known.

Below are given a few hypothetical computations. In some cases both the fuel consumption and the fuel cost are assumed outright, in others an attempt has been made to determine the

fuel consumption in relation to the size of the engine. These figures are followed by a few fuel cost figures from actual practice. Additional data for fuel cost from actual operation will be found in the data on total operating costs at the end of this chapter.

The following table shows an estimate taken from the catalogue of a well-known manufacturer.

RELATIVE COSTS OF FUEL WITH DIFFERENT TYPES OF ENGINES

Type of Engine	Class of Fuel	Price of Fuel	Fuel Consumed per B. H. P. per hour	Cost in cents per B. H. P. per hour	Cost per annum 3000 hours per 100 B. H. P.
Simple non-condensing slide valve	Bituminous Coal	$3.00 per gross ton	5 lbs.	.669	$2,007.00
Compound condensing	Bituminous Coal	$3.00	3 lbs.	.402	1,205.00
Steam Turbine	Bituminous Coal	$3.00	3 lbs.	.402	1,205.00
Oil Engine	Gasoline	14 cents per gal.	0.125 gal.	1.75	5,250.00
Oil Engine	Crude Oil	4 cents per gal.	0.10 gal.	.40	1,200.00
Gas Engine	Illuminating Gas	75 cents per 1000 cu. ft.	19 cu. ft.	1.425	4,275.00
Gas Engine	Natural Gas	20 cents per 1000 cu. ft.	13 cu. ft.	.26	780.00
Gas Engine with suction producer	Anthracite Coal	$3.00 per gross ton	1 lb.	.134	402.00
Gas Engine with suction producer	Coke	$3.00 per gross ton	1.25 lb.	.167	502.00

The next table, compiled by J. I. Wile, is similar to the above, except that the assumptions differ somewhat. A comparison of the corresponding items in the two tables serves to show the variation in the final results arrived at by computations of this kind.

STATISTICS OF FUEL CONSUMPTION AND COST PER ANNUM OF PRODUCER GAS POWER AND OTHER POWERS

Type of Engine	Kind of Fuel	Price per Ton	Fuel consumption in lbs. per B. H. P. per hour	Cost in cents per B. H. P. per hour	Cost in dollars per 100 B. H. P. per hour	Cost per 100 B.H. P. per Annum 3000 hours	
Oil Engine	Gasoline	12 cents per Gallon	1 Pint	1.50	$1.50	$4,500	Gas and Oil Engines
Gas Engine	Illuminating Gas	75c per 1000 Cubic Feet	18 Cubic Feet per B. H. P.	1.35	1.35	4,050	
Gas Engine	Natural Gas	30c per 1000 Cubic Feet	13 Cubic Feet per B. H. P	.39	.39	1,170	
Simple Steam Engine	Bituminous Coal	$3.00	8 lbs.	1.2	1.20	3,600	Steam Eng.
Compound Steam Engine Non-condensing	Bituminous Coal	3.00	5	.75	.75	2,250	
Triple Expanding Steam Engine, condensing	Bituminous Coal	3.00	2	.3	.30	900	
Producer Gas Engine	Anthracite Coal	5.00	1	.25	.25	750	Prod. Gas Eng.
Producer Gas Engine	Gas Coke	3.00	1.25	.1875	.18¾	565	
Producer Gas Engine	Bituminous Coal	2.50	1.25	.1565	.1565	470	
Producer Gas Engine	Anthracite Coal	3.00	1	.15	.15	450	

C. H. Day,* in an investigation on the economy of gas producer engines, compiled the following fuel cost data for the various kinds of prime movers mentioned. The fuel consumption figures were obtained in each case by collaborating the results of a considerable number of tests. The costs per annum were then computed by assuming the cost of unit weight of the fuel. The total time of operation per year is taken at 3000 hours.

ANNUAL FUEL COSTS FOR STEAM ENGINES. COST OF COAL ASSUMED AT $3.00 PER TON OF 2000 LBS.

Type of Engine	B. H. P.	Lbs. Coal per B. H. P. per hour	Cost of Coal per B. H. P. per year
Simple, non-condensing	50	5.75	$25.90
Simple, condensing	100	4.45	20.00
Compound, condensing	200	2.74	12.70
Compound, condensing	600	1.97	8.85
Compound, condensing	1000	1.90	8.55
Triple exp. condensing	2000	1.87	8.40

The data available for *steam turbines* is not as extensive as

* Sibley College Thesis, 1905.

that for steam engines, and the range covered is not so wide as regards capacity. The following figures, also compiled by Mr. Day, show some fuel costs for prime movers of this kind. The working time has again been assumed at 3000 hours per annum.

ANNUAL FUEL COSTS FOR STEAM TURBINES. COST OF COAL ASSUMED AT $3.00 PER TON OF 2000 LBS.

Number of Tests	Average B. H. P.	Lbs. Coal per B. H. P.-hour	Cost of Coal per B. H. P. per year
27	616	1.740	$7.83
10	1085	1.735	7.80
5	1359	1.725	7.75
25	1739	1.655	7.44

The computations made on *illuminating* and *natural gas* engines show the following results. Time of operation per year 3000 hours, cost of illuminating gas 75 cents per 1000 cu. ft., cost of natural gas 50 cents per 1000 cu. ft. The latter assumption is high, since in many localities natural gas is sold at 30 or even 20 cents per 1000 cu. ft. It is a simple matter, however, to reduce the figures in the table in the corresponding ratio.

ANNUAL FUEL COST FOR ILLUMINATING AND NATURAL GAS ENGINES. COST OF ILLUMINATING GAS ASSUMED AT 75 CENTS, THAT OF NATURAL GAS AT 50 CENTS, PER 1000 CU. FT.

B. H. P.	Cu. ft. per B. H. P.-hour		Annual Cost of Gas per B. H. P. per year of 3000 hours	
	Illuminating Gas	Natural Gas	Illuminating Gas Engines	Natural Gas Engines
10	22.0	–	$49.50	$24.00
20	21.5	16.0	48.30	23.50
30	21.0	15.5	47.20	23.30
40	20.5	15.0	46.00	22.60
50	20.0	14.5	44.80	21.80
75	19.1	13.8	42.80	20.80
100	18.3	13.0	41.10	19.60
200	—	11.4	—	17.10
300	—	10.2	—	15.40
400	—	9.7	—	14.60
500	—	9.4	—	14.10

For *producer gas* engines Day found the following figures for the consumption of gas and of coal, from which the fuel costs per year of 3000 hours are computed, assuming coal to cost $4 per ton of 2000 pounds.

ANNUAL FUEL COST OF PRODUCER GAS ENGINES

B. H. P.	Cu. ft. of Gas per B. H. P.-hour	Lbs. of Coal per B. H. P.-hour	Annual Cost of Coal per B. H. P.
50	105	1.45	$8.70
100	96	1.35	8.10
150	89	1.23	7.33
200	83	1.17	7.03
250	79	1.12	6.72
300	76	1.08	6.48
400	73	1.05	6.30
500	72	1.03	6.18

The cost of the gas for *blast furnace* gas engines has in many computations been neglected on the assumption that this gas, if not used in engines, is a mere waste product. Lately, however, it has come to be recognized that some value must be assigned to this fuel since money was expended on it in every case for cleaning it preparatory to making it fit for use in engines. The ordinary method of evaluating this gas is to compare its heat content with that of steam and to assign a value to the gas corresponding to the cost of steam. Thus the cost of the gas will in every locality vary with the cost of coal. As an example, H. Freyn * makes the following computation:

"Let us assume that the price of coal delivered into bins at the plant be $2.75 per ton, that the coal have a heat value of 13,000 B. T. U. per pound, and, further, that steam of 150 pounds boiler pressure or about 165 pounds absolute pressure be raised by burning this coal under boilers. One pound of steam will then contain 1225 B. T. U. from zero degrees Fahrenheit. Assuming feed water at 70 degrees, there would be required 1155 B. T. U. to generate 1 pound of steam at 150 pounds boiler pressure. In a boiler plant having 65 per cent efficiency, 1000 pounds of coal

* H. Freyn, Available Power and Cost of Operation of a Power Station for Waste Gases from a Blast-furnace Plant. Journal Western Society of Engineers, February, 1906.

could raise $(0.65 \times 1000 \times 13{,}000) \div 1155 = 7300$ pounds of steam. The value of 1000 pounds of steam would be $2.75 \div (2 \times 7.3) = \$0.188$, or 18.8 c. To this must be added for labor and maintenance approximately 1 c. per 1000 pounds of steam, making the total value of 1000 pounds = 19.8 c. 1000 cu. ft. of blast furnace gas have a heat value of $1000 \times 90 = 90{,}000$ B. T. U. and are equivalent to $(0.65 \times 90{,}000) \div 1155 = 51$ pounds of steam, which in turn are worth $(51 \div 1000) \times 19.8 = 1$ c.

"The value of 1000 cu. ft. of blast furnace gas would, therefore, be 1 c."

L. Eberhardt, in an article in the Zeitschrift d. V. d. I,* makes a similar computation, arriving at a somewhat different result. Starting with the assumption that 1000 pounds of steam will cost from 25 to 32 c., according to the price and quality of coal, he finds that 1000 cu. ft. of cleaned blast furnace gas should have a value of from 1.43 to 1.84 c., the gas having a heating value of 102 B. T. U. per cu. ft. This is considerably higher than the value found by Freyn, which is mainly due to the lower grade of coal (11,650 B. T. U. per pound) and the higher grade of gas assumed.

To get some idea of the fuel cost of blast furnace gas power as compared with steam power, Eberhardt, in the article mentioned, gives the following tables. The original tables give the cost per B. H. P. hour, but these have been recomputed to the basis of a year of 3000 working hours, to make them directly comparable with the figures in previous tables.

Annual Fuel Cost of Blast-furnace Gas Engines. Heating Value of Coal taken at 11650 B. T. U. per lb. Heating Value of Gas taken at 102 B. T. U. per cu. ft.

Cost of coal per ton of 2000 lbs. . . . . . . dollars	2.34	2.75	3.18
Cost of 1000 lbs. of steam . . . . cents	24.8	28.4	31.8
Cost of 1000 cu. ft. of gas . . . . cents	1.43	1.64	1.84

A fair blast furnace gas engine will show the following fuel consumption:

At full load	99 cu. ft. of gas	per	B. H. P.	hour.
At $\frac{9}{10}$ load	106	"	"	"
At $\frac{8}{10}$ load	113	"	"	"
At $\frac{2}{3}$ load	122	"	"	"
At $\frac{1}{2}$ load	131	"	"	"

* June 3, 1905.

With these assumptions of gas consumption per B. H. P., and of local cost of coal, the fuel costs per B. H. P. per year of 3000 hours will then be as per the following table:

Cost of Coal per Ton	Per Cent full Load on Engine				
	100	90	80	66	50
	Annual Cost of Blast Furnace Gas per B. H. P. Dollars				
$2.34	4.26	4.56	4.87	5.25	5.63
2.75	4.85	5.18	5.55	5.97	6.42
3.18	5.46	5.83	6.23	6.72	7.20

The fuel cost of operating on *gasoline* as fuel is of course considerably higher than any of the figures above quoted. The usual assumption made in regard to gasoline engines is that they will require 1 pint of gasoline per B. H. P. hour. This corresponds to a thermal efficiency of about 18 per cent on the brake, a figure which should be reached by a fair-sized engine in good condition. For the smaller machines, however, one pint is perhaps somewhat low, and for say a 2 horse-power machine a consumption of 2 pints per B. H. P. hour is probably a safer assumption. It should also be borne in mind that the purchase price of gasoline varies somewhat with the quantity bought. The following table of fuel costs takes these various points into account.

Fuel Cost for Gasoline Engines per B. H. P. per Year of 3000 Working Hours

Size of Engine, B. H. P.	2	4	6	10	20
Cost of gasoline per gal., cents	20	18	16	14	13
Consumption of gasoline per B. H. P. hr., gallons	.25	.20	.18	.15	.12
Fuel cost per B. H. P. per year, dollars	150.00	108.00	86.40	63.00	46.80

(*e*) Cost of Water for Cooling and Washing. Many estimates of total operating costs totally neglect the cost of water for cooling and washing purposes required by the plant, although in many localities this may amount to a considerable item of expense. In general terms, where water has to be brought from city mains, it pays to install cooling apparatus of some kind. Of course for small installations this may be simply a tank from

the surface of which the heat is radiated. The tank is connected at top and bottom with the water jacket of the engine, and the water circulates by convection. Where such a cooling tank cannot be placed in the immediate vicinity of the engine, and connected to the jacket by short straight pipes, a circulating pump of some kind must be used. As the size of the installation grows, cooling towers or cooling ponds have to be resorted to, unless an abundant source of clean water is available without cost, except that of pumping. Under such conditions the cost of cooling water is very materially reduced. With any kind of cooling system the cost then consists of the cost of pumping plus the cost of any clean water that must be supplied from time to time to replace that lost by evaporation. Of course it is not possible to give any definite figures for this cost item, because it depends upon the cost of water in the particular locality, upon the total amount of water circulated, and upon the kind of pumps used. A few figures for the amount of water required or in circulation per B. H. P. hour are given below.

In a producer plant an additional amount of water is required for the producers and scrubbers. The water vaporized for the producers of course is not again available. That used for the washing plant is in general subject to the same considerations as that used for cooling the engines. But where a cooling system is used for both, they should never be combined, because the requirements for clean cooling water for the engines are much more strict. The scrubber water usually carries considerable quantities of sediment and is contaminated with ammonia and sulfur compounds, so strongly in some cases as to give it a very noxious odor. In such cases, settling tanks and ponds, together with a considerable addition of clean water, seem to be the only remedies to recover at least a part of the water.

The approximate quantity of cooling water required may be figured from the following considerations:

One horse-power, assuming an engine efficiency of say 25 per cent at the cylinder, requires the expenditure of $\frac{2545}{.25} = 10{,}200$ B. T. U. per I. H. P. hour. The jacket loss approximates about 40 per cent of the heat expended, that is, in this case the jacket water must carry off $.40 \times 10{,}200 = 4080$ B. T. U. per hour.

Assuming the inlet temperature at 70 degrees, the outlet at 150 degrees, we find the number of pounds of water required = approx. $\frac{4080}{150-70} = 51$ pounds = say $6\frac{1}{2}$ gallons per I. H. P. hour at full load.

The following figures from various sources show how this estimate agrees with others and with some data from actual practice.

W. Heym, in the *Gasmotorentechnik*,* states that a producer plant requires per horse-power hour 6.5 gallons for the engine, 4 gallons for the scrubber, and .13 gallons for the producer vaporizer.

Freyn, in the paper already mentioned, makes the following estimates for a blast furnace gas plant of 10,500 B. H. P. Theissen washers are used to clean the gas.

GALLONS OF COOLING WATER REQUIRED

Load on Engines	For the Engines per B. H. P.-hour	For the Washers per 1000 cu. ft. of gas cleaned per minute
Full	8.5	12.0
$\frac{3}{4}$	10.5	14.0
$\frac{1}{2}$	13.0	16.0

Freyn also estimates that if the plant is located near a stream of water, the cost of pumping would probably be in the neighborhood of 2 c. per 1000 gallons.

J. R. Bibbins† on a 51-hour test of a 500 horse-power Westinghouse horizontal engine, reported a water consumption for the engines only of 9.4 gallons per B. H. P. hour at full load. The same authority reports a figure of 5.65 gallons per B. H. P. hour‡ found on similiar engines in another plant.

In conclusion it should be said that the water-consumption of gas engines depends somewhat upon the size of the cylinder. Thus a single-acting cylinder of very large diameter usually

* November, 1907.

† J. R. Bibbins, Proceedings A. S. M. E., Mid-November, 1907.

‡ J. R. Bibbins, Gas Driven Electric Power Station, Proc. of the Eng. Soc. of W. Pa.

requires more cooling water than two single-acting or one double-acting cylinder of the same capacity. Skilful attendance also is a considerable factor in the amount of cooling water used, a point which is very often neglected.

(*f*) OIL AND WASTE. The consumption of lubricating oil per B. H. P. hour depends upon the size of the engine and, as in the case of cooling water, largely also upon the care of attendants. It is also true that new machines may for a time require two or three times the ordinary amount of oil until all the parts have become adjusted to their service. In large and medium sized plants it is usual to employ a gravity or circulating system of oiling, in which case the oil is recovered, filtered, and again used. In such systems the cost of oil is a small item and consists mostly of the cost of oil necessary to replace that unavoidably lost. Of course none of the cylinder oil used is recovered.

Güldner estimates that the consumption of lubricating oil usually varies from .006 to .008 pints of oil per B. H. P.-hour, and that under favorable conditions .003 pints per B. H. P.-hour may be reached. L. L. Brewer finds that the consumption for the large engines quoted in the table, page 552, varies from .0045 to .0055 pints per B. H. P.-hour, and that in double-acting two-cycle engines the consumption may be as low as .0035 pints.

The following table compiled by J. R. Bibbins and published in the paper before the Society of Engineers of West Pennsylvania, already quoted, shows the actual oil consumption in a plant containing two horizontal Westinghouse engines of 500 horse-power each, direct connected to 300 K. W. generators. The figures cover a period of four months and the results should therefore be very reliable. The system of oiling used is of the continuous circulating and filtering type. The consumption amounts to .00202 pints of cylinder oil and .00286 pints of engine oil per horse-power hour, which agrees well with the figures given by Brewer.

Oil Consumption

4 months ending Sunday, May 23, 1906

	CYLINDER OIL*	ENGINE OIL†
Kind	Imperial Cylinder Oil	Special Gas Engine Oil
Maker	Union Petroleum Co., Philadelphia	
Price (barrel lots)	32 cents per Gal.	18 cents per Gal.
Quantity used	561 Gals.	765 Gals.
Quantity used per month	140¼ Gals.	191¼ Gals.
Quantity used per (opertg.) day	4.68 Gals.	6.38 Gals.
Quantity used per full day	6.07 Gals.	8.27 Gals.
Engine Hours per day	37	37
Engine H.P. Hours per day	18500	18500
Oil per Engine Hour	0.127 Gals.	0.172 Gals.
Oil per H.P. Hour	0.000253 Gals.	0.000345 Gals.
Cost per Engine Hour	4.05 cents	3.1 cents.
Cost per H.P. Hour	0.00809	0.00621
Total Cost	0.0143 cents per H.P. Hour	

The cost of cotton waste or other cleaning material is an item very hard to estimate, and in any case of little influence on the final result. It is usually combined with the cost of oil, and some figures for this combined cost will be found in the estimates of total operating costs at the end of this chapter.

(*g*) Attendance. The statement very often carelessly made with regard to small gas engine installations, that they require no waiting on, is of course not quite the truth. The fact is that a small gas or gasoline engine, if in good order, does not require, after starting, much attending except the proper handling of the lubricators. In this respect the gas power installation has an advantage over the small steam power plant, where at least one man is generally required all the time. For small natural gas or illuminating gas engines, therefore, it is possible to employ the "engineer," to use a familiar term, somewhere else at least part of the time, but naturally, as the size of engine increases, this spare time decreases, and it is doubtful if suction gas plants, no matter how small, can get along with less than one man's entire time.

The general methods of taking care of medium sized and large

*Includes crank case oil for exciter engines. Drainage from glands of main engines collected, mixed with old engine oil and used in crank case. No crank case oil purchased.

†Includes oil consumption of auxiliaries. Sufficient quantity old engine oil drawn from circulating system to supply auxiliaries replaced by fresh oil.

gas engines are perhaps not far different from those used in steam engine practice. It cannot be denied, however, that gas engines have much more opportunity, so so speak, to go wrong. That is, jacket water, lubrication, igniters, valves, etc., all need careful looking after, and any of these things, if neglected, may cause a shut down. While, therefore, steam engine attendants generally accustom themselves to the new service very quickly, it would be wrong to assume that they can in general take care of a gas engine plant without some instruction and practice. Of course the larger the plant, the more important this point becomes.

There seems to be nothing in English engineering literature to give an approximate idea of the time actually required in attending gas engines. Güldner proposes the following equations, apparently based on practical experience:

For illuminating gas, natural gas and oil engines,

$$W = .25\sqrt{N_n} \text{ hours.}$$

For suction gas plants $W = 1.25\sqrt{N_n}$ hours

where W = time of attendance required in hours per day of 10 hours and N_n = rated capacity of plant.

Thus, for instance, a 100 horse-power natural gas engine would actually require about $.25\sqrt{100} = 2.5$ hours in a 10-hour shift. Hence the attendant's time could be largely used somewhere else. On the other hand, a 100 horse-power suction gas plant would actually require $1.25\sqrt{100} = 12.5$ hours, which means that two men will have to be employed.

Where a station is made up of several smaller units and producers, the cost of attendance naturally increases. Thus if there are n units in a plant of N_n horse-power total, Güldner states that the actual time required per operating day of the plant may be expressed by

$$W = 1.25\sqrt{nN_n} \text{ hours.}$$

Thus if we take a 1200 horse-power suction gas plant made up of 4 units of 300 horse-power each,

$$W = 1.25\sqrt{4 \times 1200} = 86.5 \text{ hours,}$$

which would mean the service of 9 men in the 10-hour shift. If

this capacity had been put in one unit, the time would have been

$$W = 1.25 \sqrt{1200} = 43.5 \text{ hours},$$

which would have called for only 5 men per shift.

If in any of the above equations we introduce the hourly wage scale, we may express the cost of attendance per B. H. P. per hour. Thus if the scale should be 20 c. per hour, we would have for illuminating gas, natural gas, or oil engines, for the 10-hour shift,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times .25 \sqrt{N_n}}{10\, N_n} = \frac{.50}{\sqrt{N_n}} \text{ cents.}$$

For suction gas plants,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times 1.25 \sqrt{N_n}}{10\, N_n} = \frac{2.5}{\sqrt{N_n}} \text{ cents}$$

and for large suction gas plants of n units,

$$\text{Cost per B. H. P.-hour} = \frac{20 \times 1.25 \sqrt{nN_n}}{10\, N_n} = \frac{2.5 \sqrt{n}}{\sqrt{N_n}} \text{ cents.}$$

There is little data available to check the accuracy of these formulæ, but the two or three instances cited by various authorities for the labor cost in suction gas plants check very well with the results of the formula.

(*h*) Maintenance and Repairs. The expenditures for maintenance and repairs of a gas engine installation should not exceed 3 per cent of the purchase price of the engines and producers. There is no doubt that this item may easily run to 10 per cent and over, especially if in a producer plant insufficient attention is paid to cleaning the gas. But such conditions are not normal and should not occur in good practice.

Freyn states that the following are average figures for repair accounts for a blast furnace gas station:

2½ per cent per year of the price of the engines and electric generators.

7 per cent per year of the price of the cleaning plant.

5 per cent per year of the price of the air compressor used for starting.

2 per cent per year of the price of piping, etc.

To this may be added from 1 to 2 per cent of the cost of the building as maintenance for the building.

Total Operating Costs and Cost as Compared with other Prime Movers — As in the case of fuel costs, the information available in engineering literature on total operating costs is of two kinds. In the first kind the estimates are based entirely upon hypothetical assumptions, in the second the results are those obtained in actual operation. Again, the latter information is of course of much greater value, but except for large installations little of this class has been made public. Especially as regards small installations of gasoline, oil and illuminating gas engines there seems to be an almost absolute lack of data referring to total operating costs. For that reason, in order to get some idea of costs, it will be necessary for the time being to depend upon cost computations based on assumptions. In any concrete case these assumptions should be carefully scanned to see how they agree with actually existing conditions before forming a final idea of operating cost.

The data available from actual practice is mainly due to Mr. J. R. Bibbins * and others, of the Westinghouse Machine Company, and of course relates to Westinghouse engines. This, however, does not preclude their applicability to other similar plants. This information, combined with that from a few English plants, is about all that can be cited at the present writing.

Hypothetical computations are nearly all made on a comparative basis. Of this kind are the tables published by W. O. Webber in the *Engineering News*, August 15, 1907. It is not easy to give a concise abstract of the various factors assumed in the computation of these tables, and for that reason they are given in full, except for electric power which does not especially interest us here.

* The following is a partial list of the valuable papers published:

J. R. Bibbins, Gas Driven Electric Power Systems, Warren & Jamestown Street Railway, Eng. Soc. of W. Pa.; Gas Power for Central Stations, Am. Inst. of E. E., New York, December 18, 1903; Producer Gas Power Plant, Proc. A. S. M. E., December, 1906; Application of Gas Power to Central Station Work, National Elec. Light Assoc., Washington, D. C., June, 1907; Duty Test on Gas Power Plant, Proc. A. S. M. E., Mid-November, 1907; and A. West, Gas Power in Electric Railway Work, American Street & Interurban Railway Assoc., Philadelphia Convention, 1905.

E. E. Arnold, The High Power Internal Combustion Engine and its Fitness for Central Station Work, New England Street Railway Club, March, 1904.

Reprint from Power, December, 1903, A Gas Engine Pumping Station.

Cost of Gasoline Power

Size of plant in H.P.	2	6	10	20
Price of engine in place	$150.00	$325.00	$500.00	$750.00
Gasoline per B. H. P. per hour	⅓ gal.	¼ gal.	⅙ gal.	⅛ gal.
Cost per gallon	$0.22	$0.20	$0.19	$0.18
= cost per 3,080 hours	$451.53	$924.00	$975.13	$1,386.00
Attendance at $1 per day	308.00	308.00	308.00	308.00
Interest, 5 per cent	7.50	16.25	25.00	37.50
Depreciation, 5 per cent	7.50	16.25	25.00	37.50
Repairs, 10 per cent	15.00	32.50	50.00	75.00
Supplies, 20 per cent	30.00	65.00	100.00	150.00
Insurance, 2 per cent	3.00	6.50	10.00	15.00
Taxes, 1 per cent	1.50	3.25	5.00	7.50
Power cost	$824.03	$1,371.75	$1,498.13	$2,016.50

To these figures should be added charges on space occupied as follows:

Value of space occupied	$100.00	$150.00	$200.00	$300.00
Interest, 5 per cent	$5.00	$7.50	$10.00	$15.00
Repairs, 2 per cent	2.00	3.00	4.00	6.00
Insurance, 1 per cent	1.00	1.50	2.00	3.00
Taxes, 1 per cent	1.00	1.50	2.00	3.00
Total annual charge for space	$9.00	$13.50	$18.00	$27.00
Total cost per annum	$833.03	$1,385.25	$1,516.13	$2,043.30
Cost of 1 H.P. per annum, 10 hour basis	416.51	239.87	151.61	102.17
Cost of 1 H.P. per hour	$0.1352	$0.0780	$0.0492	$0.0331

Cost of Gas Power

$1.50 per 1000 cubic feet of gas less 20 per cent, if paid in 10 days = $1.20 net, gas 760 B. T. U.

Size of plant in H.P.	2	6	10	20
Engine cost in place	$200.00	$375.00	$550.00	$1,050.00
Gas per H.P.-hour in cubic feet	30	25	22	20
Value of gas consumed, 3080 hours	$221.76	$554.40	$843.12	$1,478.00
Attendance, $1 per day	308.00	308.00	308.00	308.00
Interest, 5 per cent	10.00	18.75	27.50	52.50
Depreciation, 5 per cent	10.00	18.75	27.50	52.50
Repairs, 10 per cent	20.00	37.50	55.00	105.00
Supplies, 20 per cent	40.00	75.00	110.00	210.00
Insurance, 2 per cent	4.00	7.50	11.00	21.00
Taxes, 1 per cent	2.00	3.75	5.50	10.50
Power cost	$615.76	$1,023.65	$1,387.62	$2,237.50
Annual charge for space	9.00	13.50	18.00	27.00
Total cost per annum	$624.76	$1,037.15	$1,405.62	$2,264.50
Cost of 1 H.P. per annum, 10 hour basis	312.38	172.86	110.56	143.22
Cost of 1 H.P. per hour	$0.1014	$0.0561	$0.0456	$003.67

COST OF STEAM POWER

Size of plant in H.P.	6	10	20
Cost of plant per H.P.	$250.00	$220.00	$200.00
Fixed charge, 14 per cent	$35.00	$30.80	$28.00
Coal per H.P.-hour, in pounds	20	15	12
Cost of coal at $5 per ton	$154.00	$103.00	$82.50
Attendance, 3080 hours	75.00	50.00	30.00
Oil, waste and supplies	15.00	10.00	6.00
Cost 1 H.P. per annum, 10-hour basis	$279.00	$194.80	$146.50
Cost of 1 H.P. per hour	$0.0906	$0.0832	$0.0475

ANNUAL COST OF POWER PER BRAKE HORSE-POWER *

B. H. P. of Unit	Steam	Gas	Gasoline
1	$600.00	$380.00	$487.50
2	500.00	312.50	416.00
3	437.50	260.00	350.00
4	375.00	220.00	300.00
5	320.00	192.50	262.50
6	280.00	172.50	240.00
7	250.00	160.00	210.00
8	230.00	152.50	182.50
9	210.00	145.00	165.00
10	195.00	140.00	152.00
12	175.00	132.50	137.50
14	165.00	126.00	122.00
16	157.50	120.00	112.50
18	150.00	116.50	107.50
20	146.00	113.00	102.00
22	140.00	110.00	98.00
24	137.50	107.50	95.00
26	133.00	105.00	92.50
28	130.00	102.50	90.00
30	127.50	102.00	87.50
35	124.00	100.00	85.00
40	120.00	98.00	82.50
50	112.50	96.00	80.00
60	105.00	94.00	78.00
70	100.00	92.00	76.00
80	95.00	90.00	74.00
90	90.50	88.00	72.00
100	86.40	86.00	70.00

Attention is called in these tables to the high allowance for repairs in the case of the gasoline and gas engine and the high coal consumption assumed for the steam engine.

Another interesting comparison is that made by H. A. Clark

* Unit costs: Coal, $5 per ton; gas $1.20 per 1000 cubic feet, at 760 B. T. U.; gasoline, $0.20 per gallon.

in a paper on the Diesel Engine.* In this case the computations are carried out for three sizes of Diesel engine, Crossley Gas Engine and Dowson producer, and high-speed compound condensing steam engine. The conditions assumed are partly as follows, the remainder being given in the table itself.

Cost of fuel delivered:

Diesel engine, crude oil at $10.90 per ton = appr. 3.8 c. per gallon.

Crossley engine, anthracite at $5.70 per ton.

Steam engine, coal at $3 per ton.

The fuel cost is based on the following assumption:

For the Diesel, the consumption has been taken in each case as the mean between full and half-load rates. These rates were actually determined by tests.

For the Crossley, 1.5 to 1.25 pounds of coal per B. H. P. hour plus an allowance for stand-by losses.

For the steam engine, 4 to 3.5 pounds of coal per B. H. P. hour for the lowest and highest powers, assuming an evaporation of 8 pounds of water per pound of coal.

The cost figures in the original table have all been transposed to dollars and cents. The computations are based on a year of 2700 working hours.

In the discussion on Clark's paper, the cost computations were rather severely criticised, mainly on account of the fuel costs assumed. The claim was made that both the coal consumption for the steam engine and the cost of the steam coal was assumed too high, and that the cost of oil at 3.8 c. per gallon could only apply to seaboard towns. As far as American conditions are concerned, the assumption regarding the steam engine would seem to be about right, while the cost of anthracite at $5.70 per ton for the gas engine is certainly not too low. On the other hand, the price of 3.8 c. per gallon for the crude oil would seem to favor the Diesel engine. It all comes to the point that applies to all computations of this kind, and that is that the results are strictly applicable only to the locality for which they are computed. Allowing for possible difference in the labor costs, however, it seems to the writer that the comparison is quite fair as between the steam and the gas engine.

* Proc. of M. E., 1903, II.

TOTAL COSTS IN POWER PRODUCTION

1. Means of production	Diesel Engine	Gas Engine	Steam Engine	Diesel Engine	Gas Engine	Steam Engine	Diesel Engine	Gas Engine	Steam Engine
2. Brake horse-power	35	35	35	80	80	80	160	160	160
3. Size of engine house, ft.	12x20x15	12x20x15	12x20x15	16x24x20	16x24x20	14x24x20	20x27x20	20x27x20	17x27x20
4. Ground for gas holder, ft.	—	10x20	—		11x24			12x26	
5. Size of boiler house and gas producer, ft.	—	8x16x12	—		10x18x15	15x46x20		10x20x16	20x48x20
6. Cost of land at $1.70 per square yard	$ 49.00	$ 107.00	$ 146.00	$ 73.00	$ 141.00	$ 194.00	$ 104.00	$ 199.00	$ 272.00
7. Cost of buildings	437.00	631.00	1115.00	681.00	925.00	1750.00	875.00	1140.00	2040.00
8. Engine foundations, etc., boiler settings	49.00	73.00	291.00	98.00	136.00	389.00	170.00	194.00	487.00
9. Engines, tanks, piping and condenser, etc.	$2890.00	1360.00	1165.00	4370.00	3080.00	2430.00	8170.00	5550.00	3890.00
10. Boiler or gas producer and accessories	—	974.00	1408.00	—	1260.00	1700.00	—	1845.00	2910.00
11. Total capital cost	3425.00	3145.00	4125.00	5222.00	5544.00	6463.00	9319.00	8928.00	9589.00
12. Capital cost per B. H. P.	97.90	90.00	117.70	65.25	69.30	80.80	58.20	55.80	59.90
ANNUAL CHARGES	$	$	$	$	$	$	$	$	$
13. Interest on capital, 4 per cent	135.20	126.50	165.50	209.00	222.00	259.00	373.00	357.00	384.00
14. Building, maintenance and depreciation, 5 per cent	21.90	31.60	55.90	34.10	43.80	87.50	46.20	57.20	104.00
15. Engines and machinery, maintenance and depreciation, 10 per cent	284.50	136.20	116.80	438.00	309.00	243.00	817.00	554.00	389.00
16. Boilers. Maintenance and depreciation, 12 per cent	—	—	170.50	—	—	204.00	—	—	350.00
17. Gas producer and holder. Depreciation, 5 per cent	—	48.70	—	—	63.20	—	—	92.30	—
18. Fuel	231.00	438.00	632.00	477.00	1000.00	1265.00	904.00	1670.00	2140.00
19. Lubricant, waste, etc.	97.40	97.40	97.40	170.50	170.50	170.50	268.00	268.00	268.00
20. Wages, removal of ash, etc.	365.00	438.00	642.00	365.00	462.00	535.00	426.00	535.00	680.00
21. Total	1133.00	1316.40	1700.00	1693.60	2270.50	2764.00	2834.20	3533.50	4315.00
22. Annual cost per B. H. P.	32.40	37.60	48.70	21.19	28.38	34.55	71.72	22.09	26.30
23. Cost per B. H. P. hour, cents	1.08	1.38	1.78	.78	1.04	1.26	.64	.80	.98

As a final example of this kind of hypothetical computation, the following table is given. The figures are due to Mr. L. G. Findlay of the De La Vergne Machine Company of New York. They were presented at the Dayton meeting of the Ohio Society of Mechanical, Electrical, and Steam Engineers, and were published in Power, July, 1907. The table is very complete and would repay careful study. Especially interesting is the effect of the load factor on operating costs. This factor is here defined as the average load on the plant divided by the rated capacity of the plant.

In Clark's tables above no mention is made of the load factor and it is evidently assumed to be 100 per cent. That, however, is a condition existing and maintained in very few plants. In fact, if it were, it would be poor engineering, as it gives the plant but very little overload capacity. The marked effect that a decrease of the load factor from 100 per cent to 50 per cent has upon operating costs is very clearly brought out in the table. In the tables given below for actual cost data, the load factor is in most cases even lower than this.

In the discussion on the results of the table following its presentation, the steam men objected to the comparison between the steam and gas plants at the rated load, on account of the much greater overload capacity of the steam engine plant. Lines 36, 37, 38, and 39 of the table are computed on the assumption that 80 per cent of the steam used by the engines is available for heating, but no standing charges are made against the heating plant.

Turning next to data from actual practice, the first of the following tables was given by Mr. J. E. Dowson in a paper in the *Journal of the Institute of Electrical Engineers* April, 1904. The table refers to results obtained in two English plants; the first is equipped with Dowson generators and Crossley engines, the second with the same type of producer and Westinghouse vertical engines of comparatively small size. In spite of the higher cost of coal, the greater capacity of the Walthamstow plants brings down the wage and repair items enough to make the total works costs not far different. The table only gives works cost, and it should be remembered that the results apply to English conditions.

COST OF 200 HORSE-POWER AND 1000 HORSE-POWER STEAM AND GAS PLANTS

Assuming the 1000 horse-power engine is compound condensing and the 200 horse-power is a simple automatic cut-off engine; that the coal with steam plants and bituminous coal producer costs $2 per short ton, while anthracite is $4.50 per ton. Bituminous coal averaging 11,000 B. T. U. per pound and the anthracite 13,000 B. T. U., lower heat values.

	Steam				Producer Gas							
					Bituminous				Anthracite			
	1000 H. P.		200 H. P.		1000 H. P.		200 H. P.		1000 H. P.		200 H. P.	
Load Factor	Full	50 per cent	Full	50 per cent	Full	50 per cent	Full	50 per cent	Full	50 per cent	Full	50 per cent
1 Cost of engine per rated horse-power		18.00		14.00		40.00		44.00		40.00		44.00
2 Cost of piping per rated horse-power		6.00		5.00		3.50		3.00		3.50		3.00
3 Cost of condensers per rated horse-power		3.00		. .		. .		. .		. .		. .
4 Cost of pumps, etc., per rated horse-power		0.50		. .		. .		. .		. .		. .
5 Cost of air starting app. per rated horse-power		. .		. .		1.00		1.50		1.00		1.50
6 Total of engine plant		27.50		19.00		44.50		48.50		44.50		48.50
7 Depreciation, 4 per cent on total		1.10		0.76		1.78		1.94		1.78		1.94
8 Repairs, 2 per cent on total		0.55		0.38		0.89		0.97		0.89		0.97
9 Interest, 5 per cent on total		1.37		0.95		2.22		2.42		2.22		2.42
10 Total of lines 7, 8, 9		3.02		2.09		4.89		5.33		4.89		5.33
11 Boilers per rated horse-power		10.00		12.00		. .				. .		
12 Producers per rated horse-power		. .		. .		15.00		18.50		10.00		14.00
13 Chimneys and breeching per rated horse-power		7.00		3.50		. .		. .		. .		. .
14 Heaters, feed-pumps per rated horse-power		1.50		1.50		. .		. .		. .		. .
15 Auto. stokers or coal machinery per rated horse-power		5.00		. .		2.00		. .		2.00		. .
16 Total of fuel plant		23.50		17.00		17.00		18.50		12.00		14.00
17 Depreciation, 8 per cent on total cost		1.88		1.36		1.36		1.48		0.96		1.12
18 Repairs, 3 per cent on total cost		0.70		0.51		0.51		0.55		0.36		0.42

19	Interest, 5 per cent. on total cost			1.17		0.85		0.85		0.92		0.60		0.70
20	Total of lines 17, 18, 19			3.75		2.72		2.72		2.95		1.92		2.24
21	Coal used in B. H. P.-hr. in lb.		2.00	2.20	4.75	6.30	1.10	1.50	1.10	1.50	0.90	1.40	0.90	1.40
22	Cost of coal per B. H. P. per day of 24 hours	Cents.	4.80	5.28	11.40	16.22	2.64	3.60	2.64	3.60	4.86	7.56	4.86	7.56
23	Attendance of engine B. H. P. per day of 24 hours	Cents.	1.20	2.40	3.00	6.00	1.05	2.10	..	..	1.05	2.10	..	..
24	Attendance of fuel plant per day of 24 hours	Cents.	0.65	1.30	1.75	3.50	0.75	1.50	3.75	7.50	0.70	1.40	3.75	7.50
25	Oil, waste and supplies	Cents.	0.50	0.80	0.50	0.80	0.50	0.70	0.50	0.70	0.50	0.70	0.50	0.70
26	Total daily expense per B. H. P. per 24 hour day	Cents.	7.15	9.78	16.65	26.52	4.94	7.90	6.89	11.80	7.11	11.76	9.11	15.76
27	Yearly oper. expenses 309 days per B. H. P., dollars		22.09	30.22	51.45	81.95	15.26	24.41	21.29	36.46	21.97	36.33	28.15	48.70
28	Cost of coal per B. H. P. per day of 10¼ hours	Cents.	2.05	2.25	4.86	6.46	1.13	1.54	1.13	1.54	2.30	3.58	2.30	3.58
29	Attendance of engine B. H. P. per day of 10¼ hours	Cents.	0.70	1.40	1.50	3.00	0.40	0.80	..	..	0.40	0.80	..	..
30	Attendance of fuel plant per day of 10¼ hours	Cents.	0.28	0.56	0.88	1.75	0.45	0.90	2.00	4.00	0.45	0.90	2.00	4.00
31	Oil, waste and supplies per day of 10¼ hours	Cents.	0.27	0.49	0.26	0.49	0.27	0.46	0.27	0.46	0.27	0.47	0.27	0.47
32	Total daily expenses per day of 10¼ hours	Cents.	3.30	4.70	7.50	11.70	2.25	3.70	3.40	6.00	3.42	5.75	4.57	8.05
33	Yearly oper. expenses 309 days per B. H. P., dollars		10.20	14.52	23.17	36.15	6.95	11.43	10.50	18.54	10.57	17.77	14.12	24.87
34	Yearly expenses lines 10, 20, 27 = 24-hr. day		28.86	43.76	56.26	91.57	22.87	39.63	29.57	53.02	28.78	49.95	35.72	63.84
35	Yearly expenses lines 10, 20, 33 = 10¼-hr. day		16.97	28.06	27.98	45.77	14.56	26.55	18.78	35.10	17.38	31.39	21.79	40.01
36	Cost of power, all exhaust steam used for heating		24 -hr. day.		37.61	61.25								
37	Cost of power, same as line 36 but short run		10¼-hr. day.		19.07	30.68								
38	Cost of power, 50 per cent exhaust steam for heating		24 -hr. day.		42.41	69.56								
39	Cost of power, same as line 38 but short run		10¼-hr. day.		21.11	34.90								
	Total cost, lines 6 + 16 for entire plant		$51,000.00		$7,200.00		$61,500.00		$13,400.00		$56,500.00		$12,500.00	
	Water evaporated per lb. coal, at 212°			10.00		8.00								

The second and third tables following are due to Mr. J. R. Bibbins of the Westinghouse Machine Company and were published in a paper on the Application of Gas Power to Central Station Work, read before the National Electric Light Association at Washington, June, 1907. Both of the stations mentioned use natural gas for fuel, but the Bradford plant is equipped with rather small Westinghouse vertical engines, while the Warren and Jamestown station contains two large direct connected units of 500 horse-power each. In both cases the load factor is considerably less than 50 per cent; in the case of the Bradford plant it is only about 19 per cent. The figures for the latter plant are remarkable for the length of time covered, 8½ years.

WORKS COST OF OPERATING GAS PLANTS

	Plant	
	Midland Railway Co., Leicester	Urban District Station, Walthamstow
Engine:		
Capacity tested	300	1500
Number	6	7
Make	Crossley	Westinghouse
Length of run	5 months	12 months
Load factor	—	15.25
K. W.-hour generated	200497	659756
Remarks — service	Arc and incandescent lighting, Dowson gas.	Arc and incandescent lighting, Dowson gas.
Duty of plant: *		
Lbs. per K. W.-hour	3	2
Lbs. per B. H. P.	2.25	1.5
Cost:		
Coal per ton	$3.75	$6.50
Fuel per K. W.-hour	.123c.	.17c.
Oil, waste, water	.020c.	.095c.
Wages	.292c.	.015c.
Total works cost power, cents per K. W.-hour	.473c.	.445c.
Authority:	R. M. DEELY, *Supt. Locomotives.*	F. A. WILKINSON, *Elec. Engineer in charge.*

WORKS COST OF POWER — GAS DRIVEN CENTRAL STATION
BRADFORD ELECTRIC LIGHT & POWER CO., BRADFORD, PA.

Station capacity (five engines)	470 kilowatts
Number of years in operation	8.5
Average station load factor ('03–'06)	19.09 per cent
Average gas consumption per kilowatt-hour ('03–'06)	25.5 cu. ft.
Average heat efficiency at switchboard	14.14 per cent

* Including "stand-by" losses.

Works Cost of Power	Dollars per year. Average 8 years	Cents per Kilowatt-hour. Average 4 years
Fuel, gas, including heating	$3,121	0.307
Station wages	3,068	0.410
Oil, waste and supplies	454	0.054
Running repairs (total)	848	0.085
Running repairs gas engines	285	0.034
Total operating cost	$7,776	0.890

Engine repairs — $0.36 per horse-power per year.
57.00 per engine per year.
.75 per cent on investment.
Fuel — Bradford natural gas.

WORKS COSTS — GAS POWER RAILWAY PLANT

WARREN & JAMESTOWN STREET RAILWAY COMPANY

Capacity of plant (two units)	600 kilowatts
Average time operated per day	18.5 hours
Average output per day	4115 kilowatt hours
Average load, per cent rating	37 per cent

Works Cost of Power	Dollars per day	Cents per Kilowatt-hour
Average First Four Months, 1906:		
Fuel gas	$12.97	0.315
Wages	12.37	0.300
Oil	2.63	0.064
Repairs and miscellaneous supplies	3.29	0.080
Total	$31.26	0.759

Fuel — "Bradford Sand" natural gas.

In conclusion, to make the figures in the above tables readily comparable among themselves, the cost data has been recomputed to the basis of cost in cents per B. H. P. hour in all cases.

Table I gives the total operating costs for small and medium sized plants as computed from the data of Webber & Clark. The load factor is in all cases 100 per cent. The difference in the cost of steam power arrived at by the two authorities is remarkable, Clark's figures being less than 50 per cent those of Webber. The reason for this is found in two directions, different assumptions as to cost of coal, $5 per ton as against $3, and different assumptions as to the coal consumption per horse-power. Webber

assumes as high as 20 pounds of coal per horse-power for a 6 horse-power plant, and charges a 20 horse-power plant up with 12 pounds per horse-power-hour. In fact Webber's figures are very liberal throughout for all three kinds of power.

In Table II, the works and total operating costs per B. H. P.-hour are shown side by side as computed from Findlay's table.

TABLE I

Total Operating Costs per B. H. P.-hour
Small and Medium Sized Plants

Kind of Power	Kind of Fuel	Load Factor	Working hours per year	Size of Plant: 5	10	16	20	24	30	35	50	80	100	160	Authority
				Cost in cents per B. H. P.-hour											
Steam Engine.	Coal	100	3080	13.9	6.3	5.1	4.7	4.5	4.1	4.0	3.7	3.1	2.8		Webber
Gas Engine ..	Ill. Gas	100	3080	6.2	4.6	3.9	3.6	3.5	3.3	3.2	3.1	2.9	2.8		Webber
Gas Engine ..	Gasoline	100	3080	8.5	4.9	3.7	3.3	3.1	2.8	2.7	2.6	2.4	2.3		Webber
Steam Engine.	Coal	100	2700							1.78		1.26		.98	Clark
Gas Engine ..	Prod. Gas	100	2700							1.38		1.04		.80	Clark
Oil Engine, Diesel	Crude Oil	100	2700							1.08		.78		.64	Clark

TABLE II

Total Operating and Works Cost per B. H. P.-hour.
Computed from Table by Mr. L. G. Findlay.
Works costs in brackets.

Hours in Operation per day	Load Factor	Size of Plant B. H. P.	Kind of Power: Steam	Bituminous Producer Gas	Anthracite Producer Gas
			Cents per B. H. P.-hour		
24	100	1000	.39 (.30)	.31 (.21)	.39 (.30)
24	50	1000	.69 (.41)	.54 (.33)	.67 (.49)
10.25	100	1000	.53 (.33)	.46 (.22)	.55 (.33)
10.25	50	1000	.88 (.47)	.84 (.36)	.99 (.56)
24	100	200	.76 (.69)	.40 (.29)	.43 (.38)
24	50	200	1.23 (1.10)	.71 (.49)	.86 (.66)
10.25	100	200	.88 (.73)	.59 (.33)	.69 (.45)
10.25	50	200	1.44 (1.15)	1.11 (.59)	1.26 (.78)

The data from the Bradford and Jamestown plants and from the two English plants shows the following results for the works cost per B. H. P.-hour.

Fuel	Load Factor	B. H. P. rated	Works Cost per B. H. P.-hour cents	Plant
Natural gas	19.09	640	.67	Bradford
Natural gas	37.0	800	.57	Jamestown
Dowson gas	—	400	.35	Midland Railway
Dowson gas	15.25	2000	.33	Walthamstow

From these figures, and those of Table II, the conclusion seems justified that it should be easily possible to produce one brake horse-power in a fair sized plant, say not under 200 horse-power, for a works cost of from .5 to .75 cents per hour, depending upon the conditions involved. This excludes fixed charges and assumes a load factor in the neighborhood of 50 per cent.

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